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REVIEW OF ENGINEERING PROGRESS ABROAD

VOL. V. No. 9 SEPTEMBER, 1944

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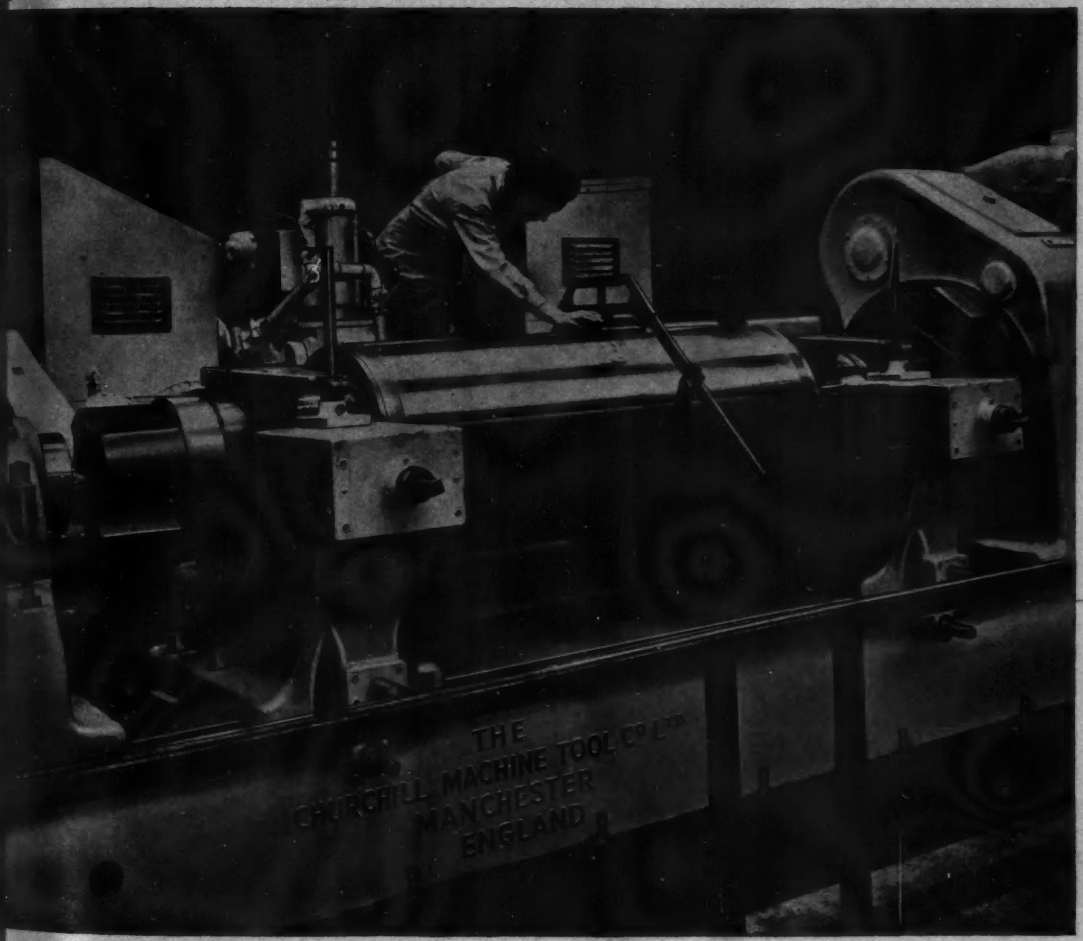

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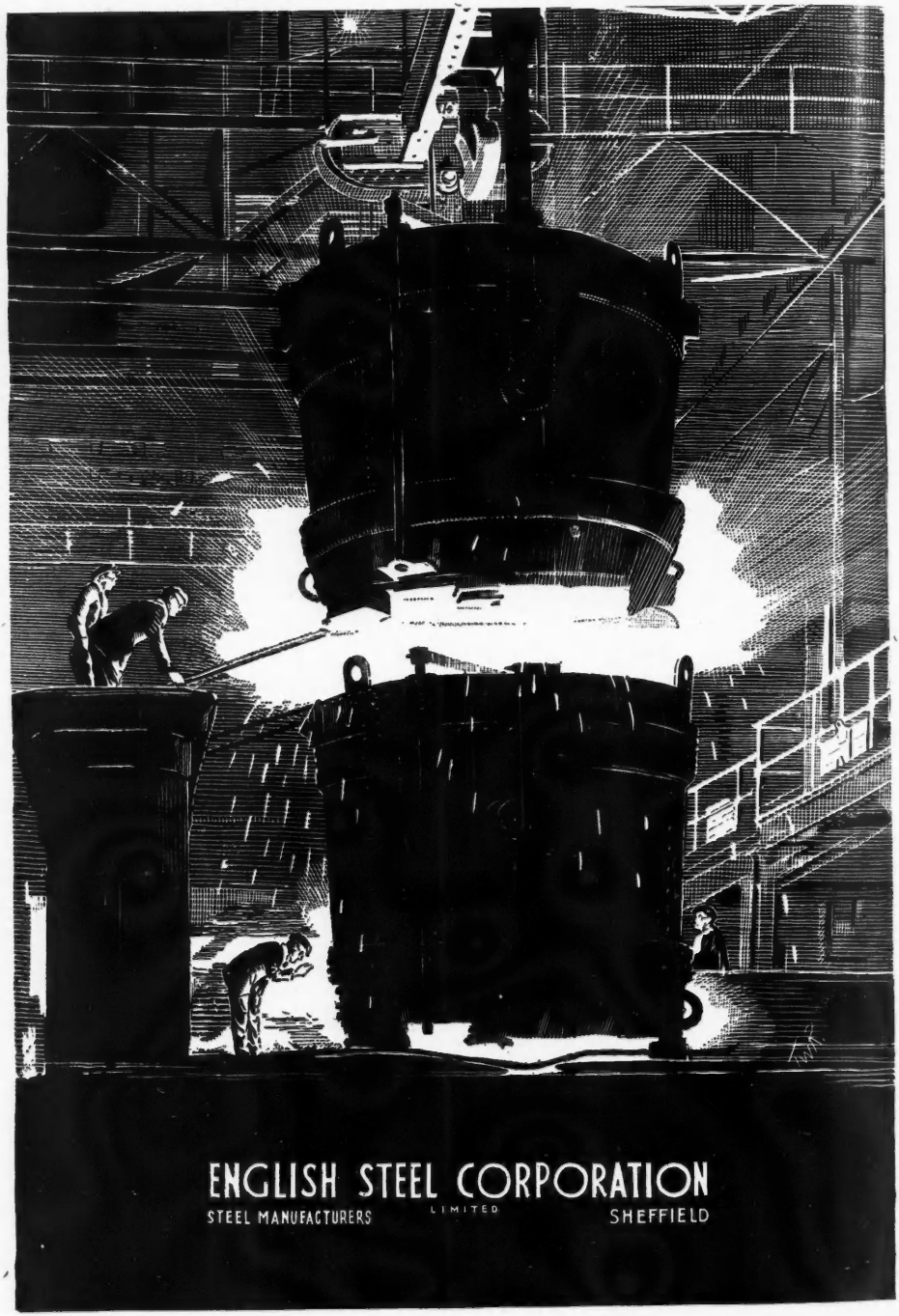
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HOW END-COIL DESIGN AFFECTS TENSION SPRINGS

By A. M. WAHL, Westinghouse Research Laboratories. (From *Machine Design*, Vol. 16, No. 7, July, 1944, pp. 107-112).

It is customary practice in spring design to use somewhat lower working stresses for helical tension springs than those permissible for compression springs of the same material and wire size. The reasons why this is necessary are somewhat obscure. Analysis indicates, however, that the practice is justified when all factors are considered. The present article will discuss these from the standpoint of fundamentals of strength of materials(1).

In the usual calculation of helical tension springs it is assumed that the load is applied axially, stresses being calculated in the body of the spring in the same manner as is done for compression springs. However, in forming the end loops the curvature at points such as A and A' (Fig. 1) may be considerably greater than the curvature of the bar in the body of the spring. This results in additional stress concentration effects which may be high, particularly if the ends are carelessly made. As a result, failure of such springs usually occurs in the bend at points such as A or A', and the spring is materially weaker than a similar compression spring. To reduce this weakening effect to a minimum the least radius of curvature of the end loops should be kept as large as practicable.

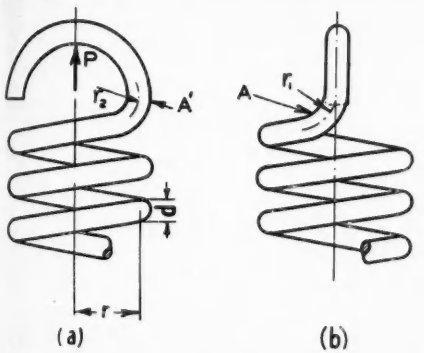

Fig. 1. **Tension spring with sharp curvature at end loop.**

A further reason why tension springs should be stressed lower than compression springs lies in the fact that it is difficult to cold set such springs properly without losing initial tension or obtaining excessive space between turns. By properly cold setting or prestressing compression springs, residual stresses are set up which tend to reduce the load stress (2). Since it is difficult to do this in tension springs, compensation for the lack of such stresses is made by reducing working stress.

Because of variations in the hook or in the method of attachment of the spring in practice, the resultant load P in Fig. 1, may be displaced from the axis by a small amount. This results in an increased stress which may require consideration. Where such springs are highly stressed it is desirable to keep this eccentricity to a minimum.

EXACT STRESS DETERMINATION IS DIFFICULT.

End Loop turned up at 60 degrees.

In the case of a tension spring with a full end loop turned up as indicated in Fig. 2, because of the complicated mathematical shape of the space curve representing the loop, an exact analysis of stress is difficult. Also the exact shape of the space curve will vary among different springs and for this reason an analysis applying to a given spring would not necessarily apply in other cases. For an end loop formed by bending up a half coil as indicated in Fig. 1 the analysis also is complicated. However, the special shape of loop shown in Fig. 3, which consists of a loop turned up at an angle of 60 deg. (Fig. 3a) can be analysed approximately without difficulty. For this reason this analysis will be carried out primarily to throw some light on the stress conditions in similar end turns.

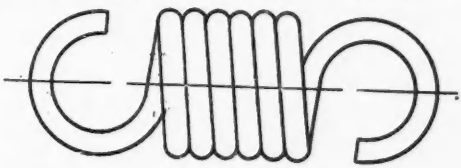

Fig. 2. **Spring with full end loop turned up.**

For a loop turned up at 60 deg. as shown in Fig. 3 the radius of curvature at all points is equal to the coil radius r. Since there are no sharp bends in such a loop, it represents a more favourable condition from the standpoint of peak stress than the loop of Fig. 1, which has a sharp bend at the point where it joins the body of the spring.

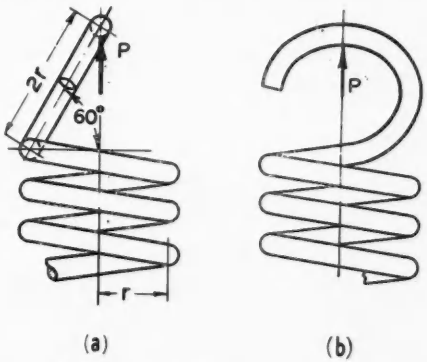

Fig. 3. **Full loop turned up at 60-degree angle.**

CONSIDERING STRESS DUE TO CURVATURE.

Analysis including Curvature effects.

Referring to Fig. 4a which represents a plan view of the loop in its plane, the end force acting in the plane of the loop will be $P \sin 60°$. Thus the bending moment on section a-a' at an angle Θ will be

$$M_b = P \sin 60° (r \sin \Theta) \quad .. \quad (1)$$

The bending stress σ_b due to the moment M_b at the inside of the coil at a' (Fig. 4a) where the maximum stress is to be expected is

$$\sigma_b = K_1 \frac{32M_b}{\pi d^3} = K_1 \frac{32Pr \sin 60° \sin \Theta}{\pi d^3} \quad .. \quad (2)$$

where d = wire diameter, r = mean coil radius and K_1 = a factor to take into account the stress augment due to curvature. An approximate expression for K_1 for a

Numbers in parentheses (1) etc., refer to references at end of article.

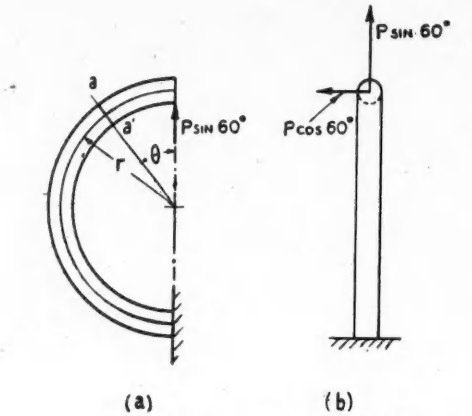

Fig. 4. **Forces acting on end loop of Fig. 3.**

circular section for values of spring index c about 3 or more is [3]

$$K_1 = \frac{4c-1}{4c-4} \quad .. \quad .. \quad (3)$$

Due to the load component $P \sin 60°$ (Fig. 4a) there will be a direct tension stress σ_t acting on section a—a'. This stress σ_t is equal to load divided by area or

$$\sigma_t = \frac{4P}{\pi d^2} \sin 60° \sin \Theta \quad .. \quad .. \quad (4)$$

The peak stress σ at a' will be the sum of the bending stress σ_b and the tension stress σ_t. Adding these stresses as given by Equations 2 and 4 and simplifying,

$$\sigma = \sigma_b + \sigma_t = \frac{32\ Pr \sin 60°}{\pi d^3}\left(K_1 + \frac{1}{4c}\right) \sin \Theta \quad .. \quad (5)$$

At section a—a' (Fig. 4a), a torsion moment M_t will act due to the load component $P \cos 60$ deg. (Fig. 4b) acting perpendicular to the plane of the loop. This torsion moment is

$$M_t = Pr \cos 60° (1 - \cos \Theta) \quad .. \quad (6)$$

Assuming that the stress augment due to curvature will be approximately the same in torsion as in bending (which is borne out by tests) the factor K_1 may be used to calculate the shear stress τ_1 due to the twisting moment M_t. This gives

$$\tau_1 = K_1 \frac{16 M_t}{\pi d^3} = K_1 \frac{16\ Pr \cos 60° (1 - \cos \Theta)}{\pi d^3} \quad (7)$$

The load component $P \cos 60°$ perpendicular to the plane of the ring (Fig. 4b) will produce a direct shear stress τ_2 at a' (Fig. 4a). This shear stress may be taken equal to that in a round cantilever acted on by an end load $P \sin 60°$ which is [4]

$$\tau_2 = 1.23 \left(\frac{4P \cos 60°}{\pi d^2}\right) \quad .. \quad .. \quad (8)$$

Adding the stresses τ_1 and τ_2 as given by eqs. 7 and 8, and simplifying, the shear stress τ at point a', Fig. 4a, becomes

$$\tau = \tau_1 + \tau_2 = \frac{16\ Pr \cos 60°}{\pi d^3}\left[K_1 (1 - \cos \Theta) + \frac{.615}{c}\right] \quad (9)$$

The bending stress σ and shearing stress τ acting on the inside of the coil at a' must be combined according to a *strength theory* to get the equivalent shearing stress.

Since in recent years there has been an increasing tendency to use the *shear-energy theory* as a basis for calculation, this theory will be used in the following. On this basis the equivalent shearing stress τ_e becomes [5]

$$\tau_e = \sqrt{\tau^2 + 1/3\ \sigma^2}$$

Using eqs. 5 and 9 in this the stress τ_e becomes, after simplification,

$$\tau_e = \tau_n \sqrt{\frac{1}{4}\left(1 - \frac{K_1 \cos \Theta}{K}\right)^2 + \left(\frac{K_2 \sin \Theta}{K}\right)^2} \quad (10)$$

where

$$\tau_n = K \frac{16\ Pr}{\pi d^3} \quad .. \quad .. \quad (10a)$$

The stress τ_n is that figured in the usual way including curvature.

$$K = \frac{4c-1}{4c-4} + \frac{.615}{c} \quad .. \quad .. \quad (11)$$

$$K_2 = K_1 + \frac{1}{4c} \quad .. \quad .. \quad (12)$$

Eq. 10 gives equivalent stress τ_e as a function of Θ. To find the angle Θ_m where τ_e has its maximum value, eq. 10 must be differentiated with respect to Θ, and the result set equal to zero. This gives

$$\cos \Theta_m = \frac{KK_1}{K_1^2 - 4K_2^2} \quad .. \quad .. \quad .. \quad (13)$$

Thus for any given index c, the values K, K_1, and K_2 may be found and from these $\cos \Theta_m$ is calculated using eq. 13. This value of Θ_m is used in eq. 10 to find the maximum equivalent stress $(\tau_e)_{max}$. The final result becomes

$$(\tau_e)_{max} = C_1 \tau_n \quad .. \quad .. \quad .. \quad (14)$$

where

$$C_1 = \sqrt{\frac{1}{4}\left(1 - \frac{K_1}{K} \cos \Theta_m\right)^2 + \left(\frac{K_2 \sin \Theta_m}{K}\right)^2} \quad (15)$$

Since τ_n is the stress in the body of the spring figured in the usual way including curvature effects, the factor C_1 represents a stress-raising or stress-concentration effect due to the end turns.

Calculated values of C_1 for various values of spring index c are given in Table I.

TABLE I.

Factor C_1 as Function of Spring Index.

Index c	Factor C_1
3	1.08
5	1.10
10	1.125
∞	1.155

The table indicates that for most practical springs which have indexes between 5 and 10 the equivalent stress in the 60-degree type of end loop shown in Fig. 3 will be around 10 to 13 per cent higher than the stress in the body of the spring. This is true even though the radius of curvature of the end loop is no smaller than that in the body of the spring. For an end loop with a sharp bend as shown in Fig. 1, an even greater weakening or stress-concentration effect may be expected. For the full loop turned up as shown in Fig. 2 conditions are probably somewhat similar to the 60 degree type of loop shown in Fig. 3.

This analysis thus indicates that some stress-concentration effect due to the end turns may be expected even when the minimum radius of curvature of the end loop is kept as large as practicable, as in the case of Fig. 2.

DESIGNING FOR STATIC LOADS.

Analysis neglecting Curvature effects.

Where static loads or loads repeated but a few times during the spring life are involved, it is common practice in design to neglect curvature effects (which result in localized peak stresses at the inside of the coil) in calculating stress. If it is assumed that the effect of the direct-shear stress under yielding conditions (due to the load component $P \cos 60°$, Fig. 4b) may be compensated for by a redistribution of stress, the stress τ_2 of eq. 8 is taken as zero. Similarly, if the direct-tension stress (due to the component $P \sin 60°$, Fig. 4a) is neglected for static loading, the stress σ_t of eq. 4 is taken as zero. This amounts to taking the factors K, K_1 and K_2 in eqs. 13 and 15 as unity. The result is the same as if c were taken as infinity, which would give a value of equivalent stress from Table I. equal to about 15 per cent above the nominal value (i.e., $C_1 = 1.155$). If static loading is assumed and if curvature, direct shear and direct tension stresses are neglected this means that the 60 degrees end turn of Fig. 3 would be about 15 per cent weaker than the body of the spring. For lower index springs this weakening effect is thus somewhat higher than that to be expected if curvature is considered.

CONSIDERATIONS FOR SHARP BENDS.

End Loop turned up sharply.

For an end loop turned up sharply as in Fig. 1b, where the radius r_1 is very small, a rough analysis of stress may be made as follows (curvature effects included): The bending moment at A' (Fig. 1a) where the sharp bends begins is Pr approximately and the section modulus is $\pi d^3/32$. Assuming a factor of stress concentration K'_2 to take into account curvature at the radius r_2 (Fig. 1a) the bending stress at A' becomes

$$\sigma_b = K'_2 \left(\frac{32Pr}{\pi d^3} \right) \quad .. \quad .. \quad .. \quad (16)$$

An approximate expression for K'_2 is [3]

$$K'_2 = \left(\frac{4c_2 - 1}{4c_2 - 4} \right) \quad .. \quad .. \quad (17)$$

where $c_2 = 2r_2/d$. (Usually the radius r_2 will differ but little from the coil radius r).

To this bending stress must be added the direct-tension stress $4P/\pi d^2$ due to the load P. Adding this to σ_b as given by eq. 16 the maximum stress σ at A' becomes

$$\sigma = K'_2 \left(\frac{32Pr}{\pi d^3} \right) + \frac{4P}{\pi d^2} \quad .. \quad .. \quad (18)$$

Taking the shear-energy theory as a basis, the equivalent shear stress τ_e at A' will be 57.7 per cent of the bending stress σ. Substituting this value in eq. 18 and simplifying, the equivalent stress at A' becomes

$$\tau_e = \tau_n \left[\frac{1.15 \quad K'_2 + \left(\frac{1}{4c} \right)}{K} \right] \quad .. \quad (19)$$

where τ_n is given by eq. 10a and is the stress figured in the usual way including curvature. For large index springs where K'_2 and K approach unity and where $r_2 \approx r$ this equation indicates stresses around 10 to 15 per cent above the nominal stress τ_n in the coils.

If static loading is assumed and curvature, direct-tension and direct-shear effects are neglected, this is equivalent to taking $K = 1$, $K'_2 = 1$ and $c = \infty$ in eq. 19. This thus indicates an equivalent stress about 15 per cent higher than the nominal stress. In other words for the end loop of Fig. 1, a weakening effect may be expected even for static loading conditions where curvature is neglected. This conclusion may be modified, however, by consideration of the redistribution of stress when yielding of the material occurs.

Higher stresses may be expected at point A (Fig. 1b), where the spring joins the body of the coil. The stress at this point may be estimated as follows: Since the loop is assumed to be bent up sharply so that the mean radius r_1 at A is very small the stress at the inside of the bend at A will be principally torsion. In this case the stress concentration factor as taken from curved-bar theory will be given approximately by eq. 3 taking $c_1 = 2r_1/d$ instead of c. This gives for the peak shear stress at A

$$\tau = K'_1 \frac{16Pr}{\pi d^3} \quad .. \quad .. \quad .. \quad (20)$$

where K'_1 represents a stress concentration effect and is approximately for $c_1 \geqslant 3$,

$$K'_1 = \frac{4c_1 - 1}{4c_1 - 4} \quad .. \quad .. \quad .. \quad (21)$$

If r_1 is small, the stress concentration factor K'_1 will be relatively large. (Thus $c_1 = 3$ and $K'_1 = 1.37$ for $r_1 = 1.5d$). This calculation shows the desirability of avoiding a small radius of curvature such as r_1 (Fig. 1), particularly where fatigue conditions are involved. This could be done by using a properly made full loop as shown in Fig. 2.

Between points A' and A (Fig. 1) there will be a combination of bending and torsion stresses which depend on the shape of the bend as well as on the radii r_1 and r_2. Since the peaks of these bending and shearing stresses do not occur at the same point, the combination of the two presents considerable complication and will not be discussed here. For practical purposes eqs. 19 and 20 are probably sufficient.

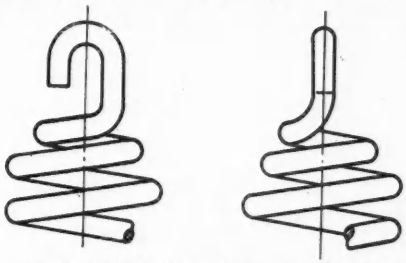

Fig. 5. **End-coil design for reduced stress.**

An interesting expedient [6] to reduce stresses in the end coils is shown in Fig. 5. In this case the diameters of the end coils are gradually reduced before the loop is formed. Then when the loop is bent up the moment arm of the load at the point where the curvature is greatest will be reduced, giving lower stresses. Such a design, though more expensive than the conventional, may be worth while where high stresses are unavoidable.

DETERMINING ACTIVE TURNS.

Effect of End Coils on Deflection.

To find the total number of active turns in a tension spring the number of turns between points where the loop begins is first determined. To this is added the deflection due to the end coils. Tests made by Sayre [7] indicate that a half coil turned up to form a loop as indicated in Fig. 1 is equivalent to 0.1-full-coil as far as deflection is concerned. Thus, if a spring has n' turns between points where the loops start, the total active turns would be $n' + 0.2$, the extra 0.2-turn being the equivalent of two loops. This conclusion may be shown analytically as follows: The half coil turned up to form a loop is equivalent to the quarter bend of Fig. 6. The deflection of this bend due to a load P is [8],

Fig. 6. **Quarter bend under load.**

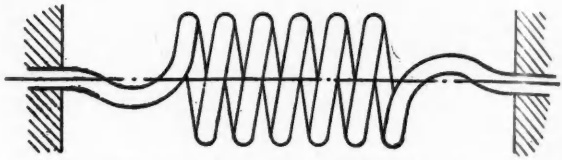

Fig. 7. **Tension-compression spring with ends coinciding with spring axis.**

$$\delta_1 = \frac{\pi}{4}\frac{Pr^3}{EI} \quad .. \quad .. \quad .. \quad (22)$$

where r = mean radius of loop (taken equal to coil radius), E = modulus of elasticity of material, I = moment of inertia of wire cross section. Taking $I = \pi d^4/64$ and the modulus of rigidity $G = E/2.6$ which is approximately true for most spring materials, by substitution in eq. 22, and simplifying the expression for δ_1 this becomes

$$\delta_1 = .096\left(\frac{64Pr^3}{Gd^4}\right) \quad .. \quad .. \quad (23)$$

This is approximately 0.1 times the deflection per turn.

For the full coil turned up as shown in Fig. 2 the experimental work of Sayre (7) indicates a deflection equal to .5-turn. In this case the total number of active coils would be $n' + 1$ where n' is the number of coils between points where the loops begin.

LARGER INDEXES REDUCE INITIAL TENSION.

Initial Tension.

The amount of initial tension which can be put into a tension spring depends primarily on the spring index $2r/d$; the higher the index the lower the initial tension values. The values of stress corresponding to practical values of initial tension listed in Table II. were published by Wallace Barnes Co. (9). These values are calculated from the usual spring formula with curvature effects being neglected. Hence the initial tension load P_1 may be figured from these values of stress using the formula

$$P_1 = \frac{\pi d^3 \tau_1}{16r} \quad .. \quad .. \quad .. \quad (24)$$

where τ_1 = initial tension stress.

As an example calculation, a tension spring has a 2-inch outside diameter and 1/4-inch wire diameter so that the index $2r/d = 7$. When wound with maximum initial tension, from Table I. the stress due to initial tension will be $r_1 = 16200$ lb./in.^{2}. From eq. 24 the initial tension load becomes

$$P_1 = \frac{\pi\,(16200)\,(.25)^3}{16\,(.87)} = 57 \text{ lb.}$$

TABLE II.

Torsional Stress Corresponding to Initial Tension.

Spring Index ($2r/d$)	Initial Tension Stress (lb./in.^{2})
3	25,000
4	22,500
5	20,000
6	18,000
7	16,200
8	14,500
9	13,000
10	11,600
11	10,600
12	9,700
13	8,800
14	7,900
15	7,000

By changes in the method of winding, values of initial tension less than this may also be obtained.

It should be noted that no appreciable deflection of the spring will occur until the external load exceeds the initial tension load. After this the slope of the load-deflection characteristic will correspond to that of a similar spring without initial tension.

REDUCES WORKING STRESSES.

Working Stresses.

It is common practice to use working stresses for tension springs around 20 or 25 per cent below those used for similar compression springs. The analysis carried out previously indicates that for a loop turned up at 60 degrees as shown in Fig. 3, a weakening effect of 10 to 15 per cent may be expected as compared with a compression spring. This case is probably somewhat comparable to the full loop of Fig. 2. Because of this weakening effect and because of the difficulty of cold setting tension springs, a reduction of working stress by 20 or 25 per cent as compared with compression springs of similar wire size and material does not appear unreasonable. Further fatigue tests may modify this conclusion, however. Where the end loop is bent up sharply as indicated in Fig. 1, if fatigue conditions are present an even greater reduction of working stress may be advisable due to stress concentration at the sharp bend.

Tension-Compression Springs.

An interesting example of a special spring which may carry both tension and compression loads is shown in Fig. 7. Such springs have proved useful in fatigue testing of machine parts where it is desired to apply alternating tension-compression loads. In forming such springs the wire or bar is brought out in a gradual loop so that the axis of the bar coincides with the spring axis as indicated. In this way stress-concentration effects due to sharp curvatures are avoided, as is important where repeated or fatigue loading is present.

Such springs are used in turbine-blade fatigue-testing machines. In this case nine springs, having the shape shown in Fig. 7, are clamped around the periphery of two circular plates. One of these plates is moved back and forth by means of a crank arrangement connected to a crosshead. The advantage of this arrangement is that any distortion or plastic flow of the test specimen will not appreciably affect the spring loads.

In designing tension-compression springs of this type it should be borne in mind that if the stress is completely reversed, the stress range will be twice that of a spring subject to pulsating (zero to maximum) stress of the same peak value.

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DOUBLE CIRCUIT FLUID BRAKES

By HANS JANDER, Berlin. (From *ATZ Automobiltechnische Zeitschrift*, Vol. 45, No. 11, June, 1942, pp. 303-311).

A REGULATION concerning passenger transport requires that fluid brakes for buses must be designed so that, in case of leakage or breakage of one pipe, the brakes of two wheels which are not on the same side of the vehicle remain in perfect operating condition. In these vehicles the foot brake must be a double circuit brake, that is there must be at least two independent circuits.

The relative setting of the brakes must remain constant over the whole braking range and must not alter during continued operation. The brake forces relative to each other, and the feel of the brake pedal, must remain the same.

Fundamentally two types of fluid brakes can be accepted :—(A) The pure air brake (Fig. 1) where at each wheel a brake cylinder is provided, or where the brake cylinders are operated by rods. (B) The air-hydraulic brake where the compressed air provides the braking force only, and the hydraulic system is used for transmitting the force.

AUTOMATIC CONTROL VALVE.

An automatic control valve, for opening the circuit to the air cylinder which is at a higher pressure, is placed between the pressure regulator and the air cylinders. When the cylinder at the higher pressure has reached a pressure of 4.5 atmos. the valve comes into operation, and the air is forced into the cylinder at the lower pressure. Thus, in case of damage to one circuit, the other remains in perfect operating condition.

The automatic control valve consists of a double acting plunger forming two valves. The air from the compressor passes between two valves, at right angles to the double acting plunger, to a spring loaded plunger which also acts as a valve. Two passages with non-return valves connect the seating of the valve to the air cylinders. The method of operation is as follows :—

In case of leakage in one circuit there is a pressure difference which causes the double acting plunger to close the circuit between the compressor and the part with the leak ; the valve on the other side of the plunger is opened, and air passes from the compressor to the cylinder at the higher pressure. When a pressure of 4.5 atmos. is reached, the spring loaded plunger valve opens, and the air passes through the non-return valve into the cylinder at the lower pressure, until here also, the pressure has reached 4.5 atmos. Both air cylinders are then charged to a pressure of 5.2 atmos., when the pressure regulator comes into operation.

The foot-brake pedal operates a double valve which is connected independently to both air cylinders. The two valves seatings are connected through separate pipes to the brake cylinders of the front and back axles.

The double brake valve is operated by means of a rocking lever pivoted at the end of a spring loaded rod. The brake pedal is pivoted to one end of the rocking lever while to the other end a cross piece is fixed which operates the double valves by means of hollow valve guides with radial holes open to the atmosphere.

When the brake pedal is depressed the springs holding the rocking lever are compressed, and simultaneously the double valves are actuated by the valve guides. If the force acting on the valve guides exceeds that on the brake pedal, the valve guides are moved to the left until the inlet valves are closed. If the force on the brake pedal is increased, the springs are further compressed and the inlet valves are opened again until a new equilibrium condition is reached.

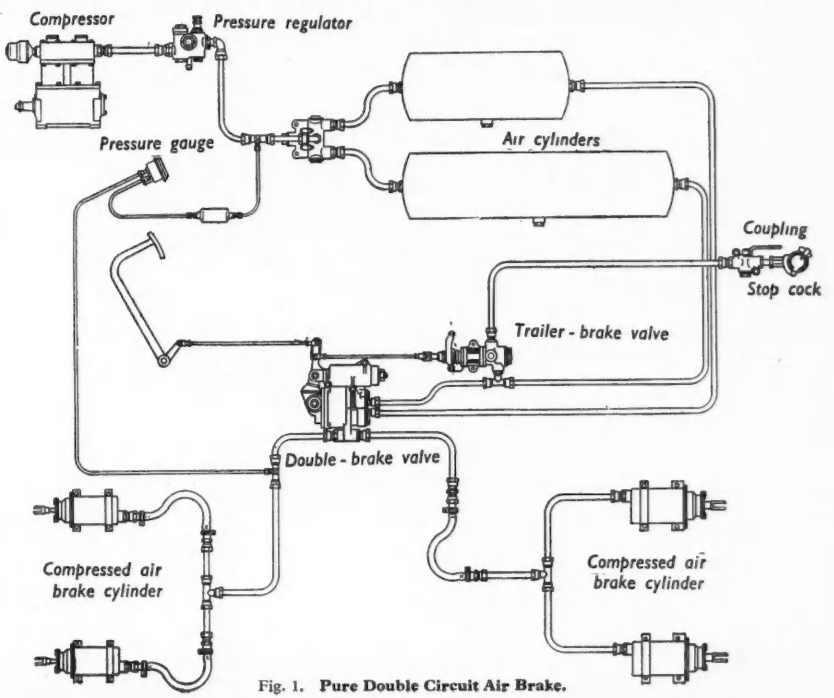


Fig. 1. **Pure Double Circuit Air Brake.**

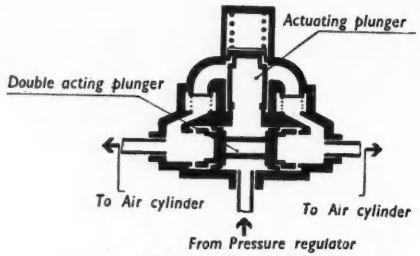


Fig. 2. **Automatic Control Valve.**

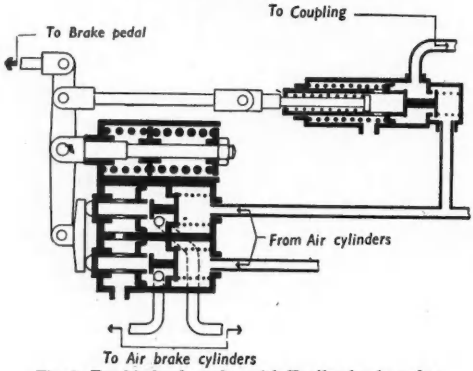


Fig. 3. **Double brake valve with Trailer brake valve.**

The brakes of a trailer are operated by means of a separate valve connected by a rod to the rocking lever. If the brake pedal is operated the tension of the spring in the cylinder is increased by the movement of the rod connected to the rocking lever. Due to the increased tension, the ratio of the forces between the plunger and the counter spring is altered, and the pressure on the plunger causes the latter to move, thus closing the inlet valve and opening the outlet valve. A pressure drop takes place by means of which the trailer brakes are operated. If a pressure drop corresponding to an increased tension of the spring in the plunger guide, takes place, the inlet valve is closed by the action of the

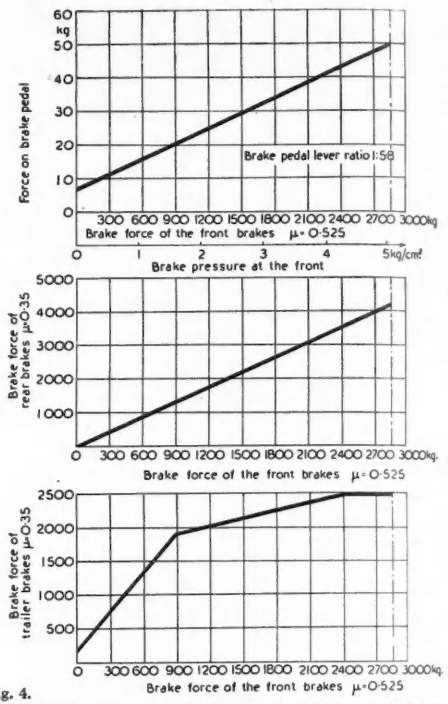

Force on brake pedal
Brake pedal lever ratio 1:5·8
Brake force of the front brakes μ=0·525
Brake pressure at the front
Brake force of rear brakes μ=0·35
Brake force of the front brakes μ=0·525
Brake force of trailer brakes μ=0·35
Brake force of the front brakes μ=0·525

Fig. 4.
Relation between various brake circuits with pure air brake.

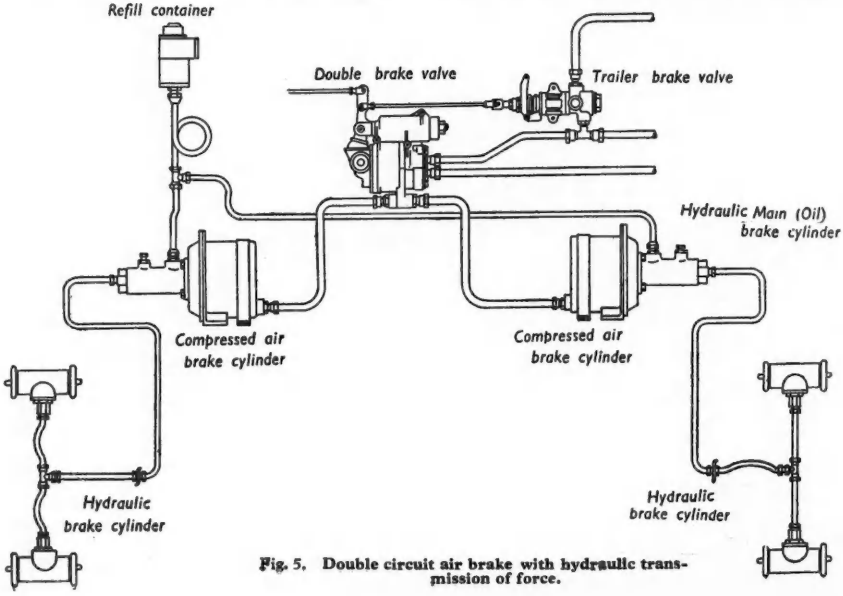


Fig. 5. Double circuit air brake with hydraulic transmission of force.

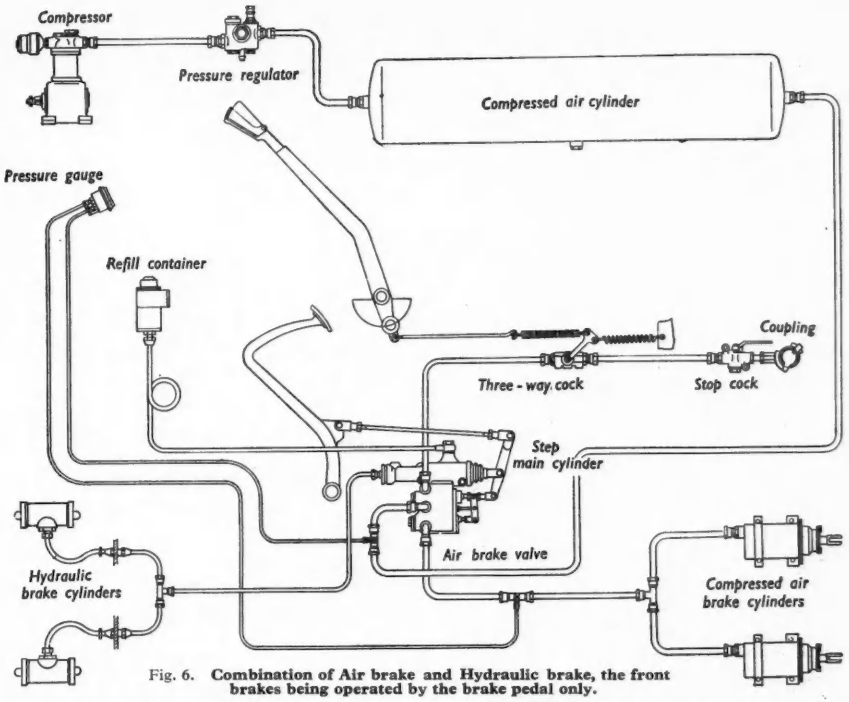


Fig. 6. **Combination of Air brake and Hydraulic brake, the front brakes being operated by the brake pedal only.**

counter spring until an equilibrium position is obtained.

By releasing the brake pedal the tension of the spring in the plunger guide is released and the counter spring causes the inlet valve to open, thus releasing the brakes.

In this arrangement the brake pressure is not determined by the force exerted on the brake pedal but by the distance through which the latter is moved.

As there is no connection between the brake pedal and the brake blocks, the travel of the former remains constant even when wear of the brake linings occurs.

Over-running of a lorry by its trailer is prevented by the counter springs of the double brake valve, these springs having different characteristics. When the pressure in the trailer pipe line has dropped by about 2 atmos., the weaker spring is compressed. Further braking of the lorry and the trailer is effected by the stronger spring only. By this arrangement for the trailer a sharp bend is obtained in the curve of brake pressure plotted against brake pedal forces, whilst the characteristic curve for the lorry brakes is a straight line.

The relation of the various brakes is shown in Fig. 4. The brake pedal force is proportional to the brake pressure on the front wheels. The characteristic of the two rear brakes is parallel to that of the front brakes. The brake pressures of the front and rear brakes are proportional, both being operated by the cross piece. Comparing the characteristics of the rear and trailer brakes, we can see that the brake pressure of the trailer brakes is larger over the whole braking range. The travel of the brake pedal and all characteristics of the brakes mentioned above remain unaltered, and are independent of wear of the linings.

In the double circuit air brake with hydraulic transmission (Fig. 5) combined air-hydraulic cylinders are provided for each axle; in other respects the two arrangements are identical both in construction and effect.

These brakes are suitable for all types of vehicles, including those with six wheels. The maximum force required at the brake pedals is 50 kg. corresponding to a travel of roughly 150 mm.

The disadvantage of both arrangements is that in the event of the air compressor cutting out, the brakes can be applied for a limited number of times only. The vehicle can be stopped with safety, although the travel of the brake pedal is increased, but operation cannot be continued and repair must be carried out on the spot.

COMBINATION OF AIR BRAKE AND HYDRAULIC BRAKE.

In this arrangement (Fig. 6) the rear brakes are operated by compressed air, while the front brakes are operated by hydraulic pressure. The front brakes are directly actuated by the brake pedal only without any servo assistance. The rear brakes are actuated by two compressed air cylinders and controlled by an operating valve which is interconnected with the brake pedal. Thus, to obtain a sufficiently small force at the brake pedal, linings with a high coefficient of friction have to be used and a "step main cylinder" has to be provided.

To this cylinder (Fig. 7) an air brake-valve is connected containing two sets of valves for the trailer and rear brakes respectively. Both sets are connected by a crosspiece mounted in the valve housing. The cross-piece is operated by a rocker pedal, the "step main cylinder", and the trailer valve are connected.

When the compressed air cuts out the hydraulic front brakes remain in good operating condition, while when the hydraulic system fails, the rear and trailer brakes remain intact.

In this case again the valves are operated through the middle pivot of the rocking lever, which becomes rigid. A strong compression spring mounted in the

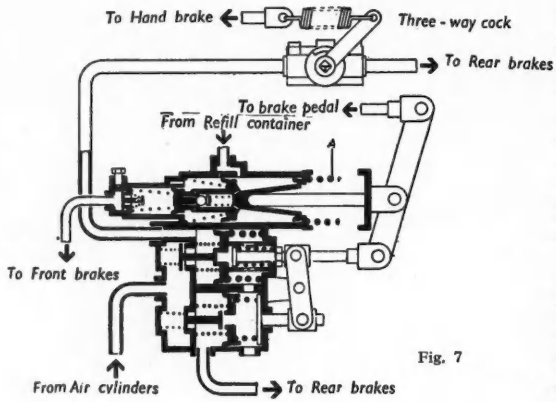


Fig. 7

casing of the hydraulic cylinder takes up the forces required to operate the valves. The length of the spring is so designed that the travel of the spring collar connected to the rod of the hydraulic plunger is about 16 mm. The spring is provided to improve the operating condition of the valves for the compressed air, in the event of the hydraulic system cutting out. The length of the spring is made to bring the compressed air into operation at the middle of the travel of the brake pedal, and not at the end. The force required on the brake pedal at the end of its travel is about 25 kg.

This arrangement brings about a considerable increase in the force required on the brake pedal, as also with the brake in good condition, the spring has to be stressed.

The lead of the trailer brakes is determined by the initial compression of the spring mounted on the trailer brake-rod. An increase in initial compression increases the lead.

The use of a "step main cylinder" results in a non-linear relationship between the forces applied at the brake pedal and the force of the hydraulic front brakes as shown in Fig. 8. The ratio of the forces is altered while changing from the low pressure step (plunger diameter 50.8 mm.) to the high pressure step. The force at the brake pedal is thus changed in the ratio of the plunger diameter. Fig. 8 shows that with uniformly increasing brake pedal force, at the instant when the step main cylinder is switched over, the force of the front brakes jumps from about 500 kg. to roughly 1250 kg. Thus with the most frequently used brake pressures of 16 to 40 atmos. it is necessary to depress the brake pedal beyond a certain point followed by a sufficient withdrawal to give the required adjustment of the force of the front brakes. To obtain the maximum retardation (60 atmos. in hydraulic system and 5.2 atmos. compressed air) a force of about 80 kg. on the brake pedal is required. The relation of the forces at the rear and trailer brakes depends on the tension of the springs operating the valves.

The force of the brake pedal, and its travel increase with wear of the brake linings; thus the relation of the forces at the rear and trailer brakes changes continously until a limiting condition is reached (dotted lines in Fig. 8).

The use of these brakes is limited to 3 and 4½ ton vehicles.

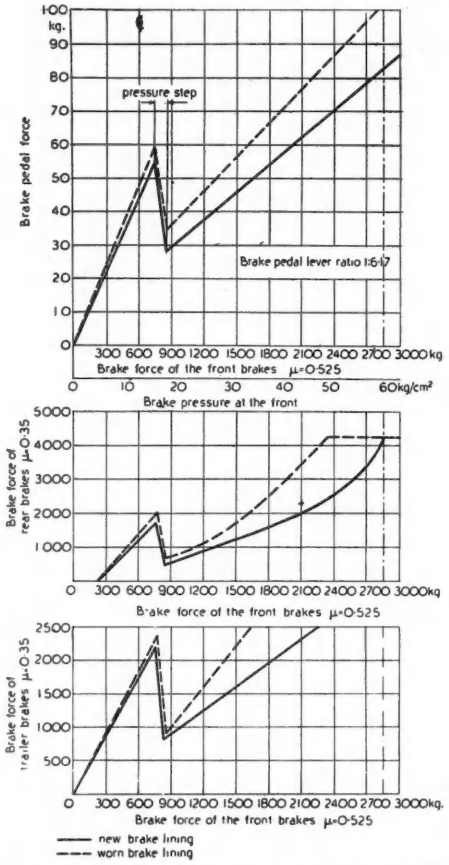

Brake pedal force
pressure step
Brake pedal lever ratio 1:6·17
Brake force of the front brakes μ=0·525
Brake pressure at the front
Brake force of rear brakes μ=0·35
Brake force of the front brakes μ=0·525
Brake force of trailer brakes μ=0·35
Brake force of the front brakes μ=0·525
new brake lining
worn brake lining

Fig. 8. Combination of Air brake and Hydraulic brake, the front brakes being operated by brake pedal only.

DOUBLE CIRCUIT BRAKE WITH REGULATING VALVE FOR FRONT, REAR, AND TRAILER BRAKES.

In contrast to the arrangements discussed so far, the three brake groups of the double circuit brake shown in Fig. 9 are operated by one regulating valve only. The front brakes are hydraulically operated by the brake pedal which is aided by means of compressed air. The rear brakes are operated by compressed air.

The increase in the brake pedal force due to the compressed air is not obtained by a separate brake cylinder, but by the regulating plunger of the brake valve which is connected to the plunger of the hydraulic brake through a rocking lever. The regulating plunger divides the valve casing (Fig. 10) into two chambers. The chamber on the left is permanently connected with the compressed air cylinder by a pipe; and to the chamber shown on the right, the rear brake cylinders are connected. The chamber on the right is opened to the atmosphere by a diaphragm outlet valve fitted in the plunger.

The diameter of the plunger in the left valve chamber is larger than that in the chamber on the right. Consequently, the effective area of the left chamber is smaller than that of the right chamber. A ratio is chosen to give a pressure decrease of 1.2 atmos. in the right valve chamber, causing the regulating plunger to move to the right and exert a force on the hydraulic plunger. The effective cross-sectional areas of the plungers of the rear brakes are operated after a pressure drop of 1.2 atmos. has taken place. The trailer brakes are also operated by the pressure drop in the valve chamber on the right and are effective after a small pressure drop has taken place.

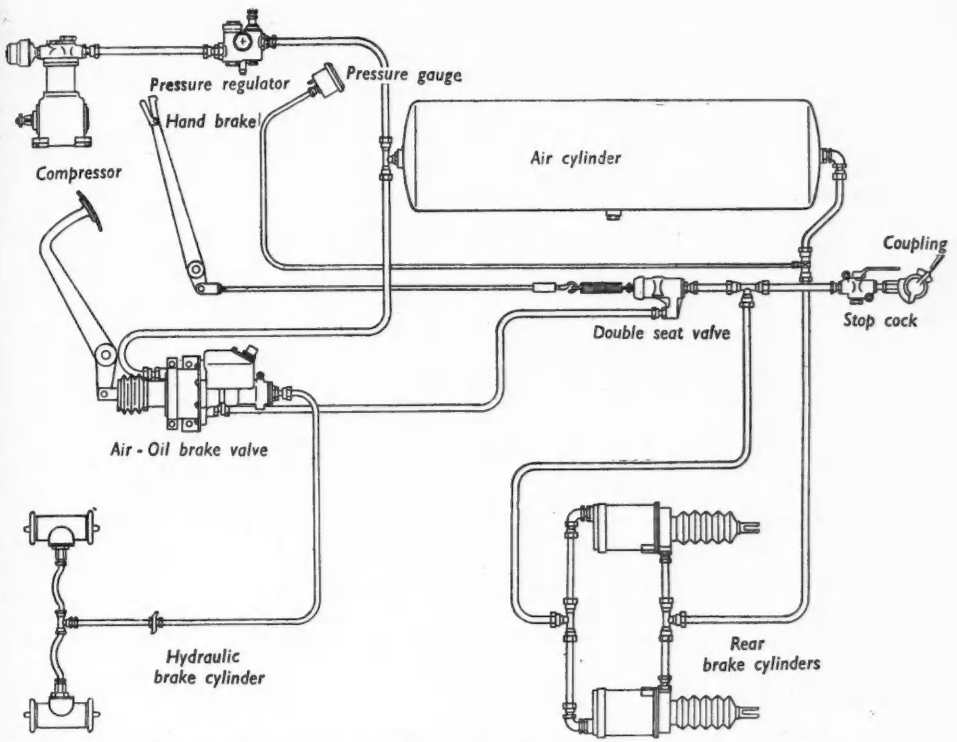


Fig. 9. **Combination of Air brake and Hydraulic brake, the front brakes being operated by the brake pedal aided by means of compressed air.**

The regulating plunger is held by a spring in the extreme left position. The object of this spring is to release the hydraulic brake when the compressed air cuts out. With the system in working order the release of the hydraulic brake is achieved, similar to that of the rear brakes, by compressed air due to the difference in cross-sectional areas of the plungers.

When the brake pedal is depressed the pressure plate in the plunger guide is moved to the right. Thus, the inlet valves in the plunger are closed and the connection between the two valve chambers is closed. During this part of the stroke of the pressure plate the diaphragm of the outlet valve is deflected, but the outlet valve is not closed. After the inlet valve has been closed a further movement of the pressure plate will open the outlet valve, thus connecting the valve chamber on the right to the atmosphere.

A pressure drop in the valve chamber on the right and in the spaces connected with it causes the trailer brakes to operate first. The rear brakes are actuated when a pressure drop of 1.2 atmos. has taken place, the regulating plunger being moved to the right by the pressure in the cylinder. The movement of the regulating plunger is transmitted by the rocking lever to the plunger of the hydraulic brake. The pivots of the rocking levers facing the pressure plate, exert a force to the left which tends to close the outlet valve and acts against the force exerted by the drivers foot. Closing of the outlet valve is prevented by the force exerted by the drivers foot on the pressure plate which first exceeds the force in the regulating plunger; the hydraulic brakes are then operated over the middle pivots of the rocking levers.

If in case of a further pressure drop the force in the regulating plunger exceeds that exerted on the pressure plate, the pivots of the rocking levers facing the pressure piece become rigid. The regulating plunger is moved to the right until the outlet valve is closed.

With this arrangement the regulating plunger of the three brake groups becomes simultaneously an auxiliary working plunger for the front brakes operated by the brake pedal. The arrangement has the following advantages :—

1. As compressed air is being used, the dimension of the hydraulic front brakes can be made sufficiently large without a " step main cylinder."
2. The three brake groups form a unit with all brake plungers at the same pressure.

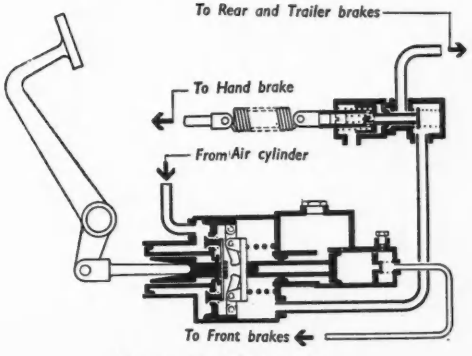


Fig. 10. **Air-Oil Brake Valve.**

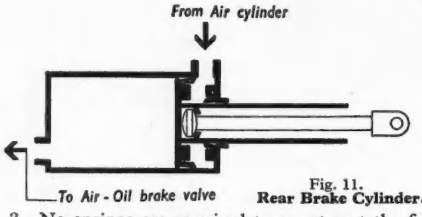


Fig. 11. **Rear Brake Cylinder.**

3. No springs are required to counteract the force exerted by the regulating plunger.

The "feel" of braking on the three brakes is directly transmitted from the hydraulic front brakes through the rocking levers.

4. The uni-axial design of the regulating valve and the primary cylinder, and the use of a regulating plunger as a working plunger, result in a large cross-sectional area of the plunger thus giving a wide field of response.

The hydraulic brakes remain in working condition when the compressed air fails. Due to the absent servo-force the force exerted by the foot is larger, but stays within limits permitting a 4½-ton lorry to be stopped with a retardation which satisfies the requirements of the authorities.

In case the hydraulic system cuts out, the rear and trailer brakes remain effective. Then a force of 50 kg. at the brake pedal will result in a retardation of more than 2.5 m. sec.^{2} in the case of a 4½-ton lorry.

The relative relationship of the brake groups is illustrated by the graphs in Fig. 12. The brake pedal force increases proportionately with the braking force of the front brakes. The aid of the hydraulic brake by compressed air results in a brake pedal force of 48 kg. at the maximum retardation (5.2 atmos. compressed air, 60 atmos. in hydraulic system). The brake force of the rear brakes is proportional to the brake force of the front brakes, and is in the same ratio over the whole braking range. The lag of the lorry brakes behind the trailer brakes is 500 kg. which corresponds to 1 atmos. The characteristics of the trailer brake force is proportional to that of the rear and front brakes of the lorry; thus the lead of the trailer brakes is constant over the whole range of brake pressures. The travel of the brake pedal changes with wear of the brake linings, this change, however, has no influence on the other brake characteristics. Because of its simple construction this double circuit brake can also be used for other vehicles.

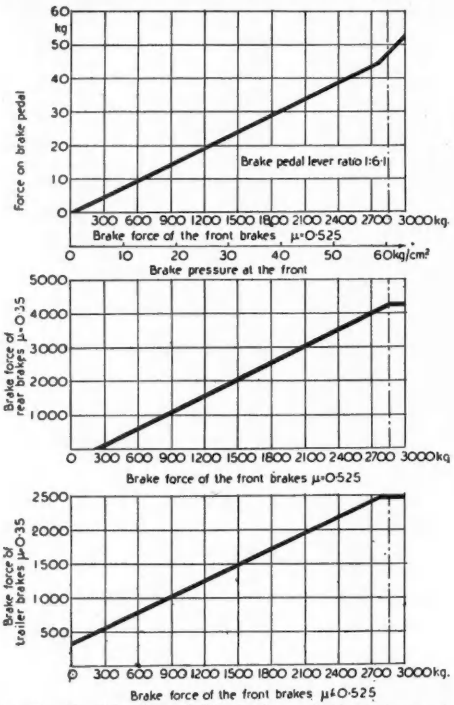
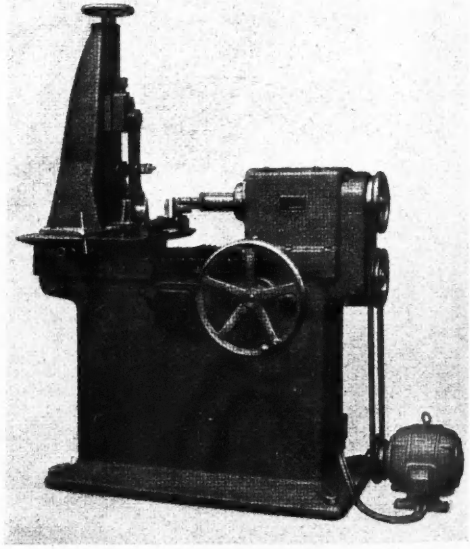


Fig. 12. **Combination of Air brake and Hydraulic brake, the front brakes being operated by the brake pedal aided by means of compressed air.**

BIG-END BORING MACHINE.

(From *Die Werkzeugmaschine*, Vol. 47, No. 6, March, 1943, p. 132).

THE horizontal boring machine for the fine boring of big-ends and big-end bushes, which is shown in the accompanying engraving, serves for finishing the big-end bearings and their faces without reclamping. The dimensions of the machine are so chosen that the longest connecting rods occurring in practice can be machined, including milling of the oil groove at the point where the bearing is split. The boring to final dimensions in one operation is achieved with the use of a setting gauge for the boring cutter. In this way the bore of the bearing is directly and exactly matched with the diameter of the crankpin, all measuring devices and attendant errors in measurement being eliminated. The cutters can be easily exchanged, so that any number of different tools can be used with the machine.



The fixture provided for clamping the big-end is noteworthy for its convenience of operation, ease of access and inspection. The big-end itself is held by a self-centering vice, but provision is made for additional clamping at a point higher up. The small-end bearing is held by a clamping device acting upon the gudgeon pin.

Both manual and automatic feed are provided. The boring spindle of the machine is hardened and is supported in adjustable anti-friction bearings furnished with forced lubrication. For the purpose of lining up and adjusting the cutter, the spindle can be rotated by hand. Four spindle speeds and three feed rates are provided.

SUPERSONIC WIND TUNNELS

By C. SEIPPEL. (From *Brown Boveri Review*, Vol. 30, No. 7/8, July/August, 1943, pp. 176–183).

THEORETICAL CONSIDERATIONS.

STREAM lines of two different compressible media in geometrically similar tunnels are similar at all points if the respective Reynolds numbers $\mathrm{Re} = \frac{wd\rho}{\eta}$ are equal. This means equal efficiencies of the two systems in which a definite ratio must exist between the dimensions d, the velocities w, the densities ρ and the viscosities η Another ratio of significance is the Mach number $M = w/a$, where a is the velocity of sound. The Mach number enables flows to be divided into two categories, the subsonic and supersonic, in which regions fundamentally different phenomena occur. To explain this, use is made of the following concept :—

A fluid in motion is guided by the wall enclosing it. We may imagine this guidance to be in the form of a large number of small impulses or blows coming from the elements of the wall. This picture is by no means entirely arbitrary if the molecular structure of matter is recalled. Every small blow produces in the stationary gas a spherical wave, which travels away from the centre of disturbance at the velocity of sound, the amplitude diminishing naturally with the distance.

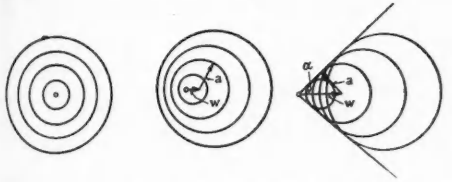


Fig. 1. **Propagation of a disturbance wave in a stream.**

Rest	Below sound	Above sound
$w = 0$	$w < a$	$w > a$
		$\sin\alpha = \frac{a}{w}$

The fundamental difference between subsonic and supersonic speed is clearly visible.

If the gas is in motion relatively to the wall element, the wave is carried along with the stream. It is no longer centred around the focus of the disturbance (Fig. 1). If the gas flows with a velocity above that of sound, the wave is no longer able to travel up against the current. It is swept along in one direction and appears only within a cone whose generatrix makes with the direction of flow the Mach angle :

$$(\sin\alpha = \frac{a}{w} = \frac{1}{M}).$$

Outside this cone it is not possible to detect any influence on the flow. Thus entirely different forms of flow and quite different flow conditions exist in the zones below and above the velocity of sound.

Special conditions exist at the boundaries of the two regions. Below the velocity of sound, dynamic pressure and lift of an aerofoil are being increased in the ratio of $1/\sqrt{1-M^2}$. Approaching the velocity of sound quite different phenomena become manifest (break away due to secondary effects). The mere fact that the ratio $1/\sqrt{1-M^2}$ becomes infinite when $M = 1$ points to an anomaly. The transition from supersonic to subsonic occurs almost always discontinuously with a compression shock which is one of the typical characteristics of supersonic flow.

Another surprising difference between supersonic and subsonic flow is to be found in the flow of fluid with friction through a cylindrical tube.

With increasing Mach number the ratio of the pressure drop to the kinetic energy increases rapidly. It becomes infinite at sound velocity, from which it follows that it is impossible for a fluid to flow through a cylindrical tube at the velocity of sound. At velocities above sound the pressure drop turns into a pressure rise. The energy loss, which of course occurs just as in the previous case, is entirely at the expense of the kinetic energy.

For imcompressible media we have the relation :

$$\Delta p = -\lambda \frac{l}{d} \frac{\rho w^2}{2}$$

For compressible media this must be replaced by :

$$\Delta p = -\frac{1 + (k-1)M^2}{1 - M^2} \lambda \frac{l}{d} \frac{\rho w^2}{2}$$

an expression which becomes infinite for $M = 1$ and which changes its sign when M becomes greater than 1.

SUPERSONIC WIND TUNNELS.

From the above considerations it follows that carrying out model tests at high velocities requires equality of both Reynolds and Mach numbers respectively for model and full scale items.

The power of the supersonic jet is quite considerable and amounts to :

$$L_w = F. \frac{\rho w^3}{2} = \lambda(M) F \frac{\rho_0 a_0^3}{2}$$

where F is the area of the test section, ρ the density in the test section and w the velocity in the test section ; ρ and a_0 denote the density and the sound velocity at the state in front of the nozzle and is a function of the Mach number, which can be taken from the diagram

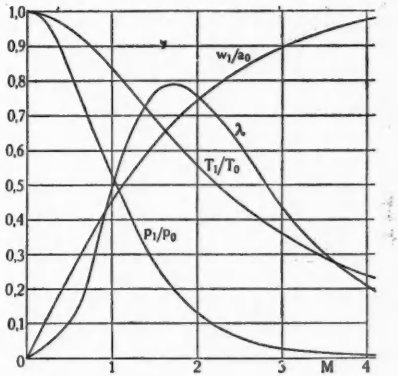


Fig. 2. **Diagram of expansion nozzle** (no loss).

p_0, T_0, a_0. Pressure, temperature and sound velocity on front of the nozzle.

p_1, T_1, w_1. Pressure, temperature and velocity of flow after the nozzle as a function of the Mach number M.

λ. Power coefficient : $\rho_1 w_0^3 = \lambda e a_0^3$.

Fig. 2. The adiabatic compression work L_k, to be supplied by a compressor in order to recompress the air from the final pressure at the nozzle, after entire loss of the kinetic energy, is greater than L_w in the ratio T_0/T (Fig. 2). For instance, for an initial condition $P_0 = 1.0$ kg./cm.^{2} abs., $T_0 = 300$ deg. K, the powers L_w and L_k per m.^{2} of test section are :

M	1	2	3
L_w	12,000	18,000	10,400 kW
L_k	14,400	32,500	29,300 kW
R_e	15	9.6	4.0×10^6

The Reynolds numbers are referred to a square test section of 1 m. length of side, whereby it should be remembered that the Reynolds numbers referred to some dimension of a model located in the test section will be correspondingly smaller. The power actually required is a fraction $1/q$ of L_k and depends on the amount of the kinetic energy which can be recuperated after the test section, and does not attain such small values in supersonic tunnels as in subsonic tunnels.

Two methods of reducing the energy requirements have been applied :—

1. By the quick discharge of an energy accumulator.
2. By reducing the pressure in continuously operating plants.

The older channels, for instance the first Göttingen installation, is of the accumulator type. A receiver of 10 m.^{3} content is evacuated and hereafter filled via the test section of 6 × 6 cm. to atmospheric pressure, during which a supersonic velocity can be maintained for 10 seconds. Other channels work with a pressure receiver which, in order to produce the desired vacuum, discharges into an ejector beyond the test section. To this type belongs a tunnel with a test section of 30 cm. diameter in England and one of 60 cm. diameter in

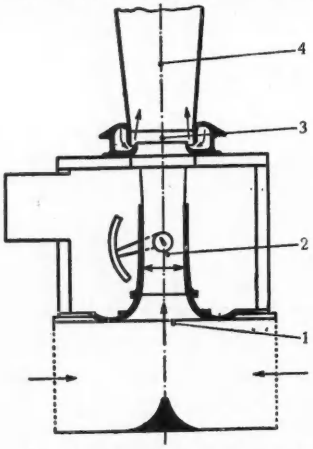


Fig. 3.
Supersonic wind tunnel with compressed air accumulator (U.S.A.).

1. Nozzle.
2. Measuring section.
3. Ejector supplied by accumulator.
4. Diffusor.

Such tunnels can only operate intermittently.

America (Fig. 3). In a Russian installation, the pressure receiver discharges through a reducing valve directly into the test nozzle. The temperature in the test section is kept under control by the injection of fuel before the nozzle. This is the only plant where the pressure and temperature in the test section reach atmospheric conditions at sea level. Clearly the new types of supersonic wind tunnels with continuously working plants are much to be preferred. The closed type consisting of a compressor, cooler, test chamber and return channel, has—compared with the open suction type—the great advantage that it is possible to regulate the level of the absolute pressure in the system by means of a pump, and hence to adjust the power requirements to the available source. A higher density with a smaller test section leads with a given power to higher Reynolds numbers than a small density with a large test section. However, it is not desirable to reduce the size of the test section below a certain minimum, especially when it is considered that in the neighbourhood of the velocity of sound the model should be small in proportion to the size of the tunnel. Moreover, the suspension of the model becomes more difficult with increasing density.

A particular advantage of the closed tunnel consists in the fact that by adjusting the pressure, the Reynolds numbers can be varied, while maintaining the same Mach number. Generally, it will never be possible to reach the large Reynolds numbers encountered in practice. All the same it is important to test whether within the attainable limits, as is generally assumed, the effect of the Reynolds number at high Mach numbers is of secondary importance. Further, the closed tunnel allows the use of gases other than air, such as monatomic or heavy gases, for which the laws of flow are different from those for air.

THE TEST SECTION.

In the test section the air flow has to be accelerated to the required velocity in the test chamber. Beyond the test chamber the kinetic energy should be converted back into pressure at as high an efficiency as possible. Fig. 4 shows such a test section with the following main parts :—

(*a*) Converging nozzle for accelerating the flow to sound velocity.
(*b*) Diverging nozzle for further acceleration of the flow above the velocity of sound.
(*c*) Measuring chamber in which the lines of flow must be as nearly as possible parallel.
(*d*) Converging part of the diffusor for reducing the velocity down to that of sound, or more exactly, as we shall see, down to a value still in the supersonic region.
(*e*) Diverging part of the diffusor for further reducing the velocity down to a value suitable for the pipe systems, and which is far below that of sound.

Fig. 4. **Test section with cover removed.**
The section comprises a convergent-divergent nozzle, a parallel walled measuring chamber of rectangular section and a convergent-divergent diffusor.

In principle, it should be possible behind the test section to reverse the sequence of conditions in the nozzle in a converging-diverging diffusor symmetrical to the nozzle. Experience shows, however, that the second neck must be greater than the first one by a definite amount, even with perfectly frictionless flow.

Let us imagine that the tunnel is put into operation in such a way that the pressure at the entrance of the test section is kept constant, whilst it is progressively lowered at the outlet. If now the second neck is narrower than the first one, the sound velocity appears there first. From that moment onward, the entire flow upstream cannot be effected by a further lowering of the pressure and it is impossible to obtain a supersonic velocity in the test section (Fig. 6a).

If on the other hand, the first neck is narrower, then the velocity of sound occurs first there (Fig. 6b). Thereupon a zone of supersound velocity is set up in the diverging nozzle, which changes to subsonic in a compression shock. With decreasing final pressure, the shock wave is displaced towards the test section and is suddenly swept down to the diverging part of the diffusor, where it becomes stationary (Fig. 6c). Further lowering of the final pressure has no effect on the test section. It is even possible to increase the back pressure slightly, the shock wave moving up towards the second neck, which, however, it must not be allowed to reach, as it would then spring over into the nozzle.

The sudden compression at the transition point involves an increase in entropy. The state of the gas after this shock is represented by another isentropic line which may be determined from the fact that if the

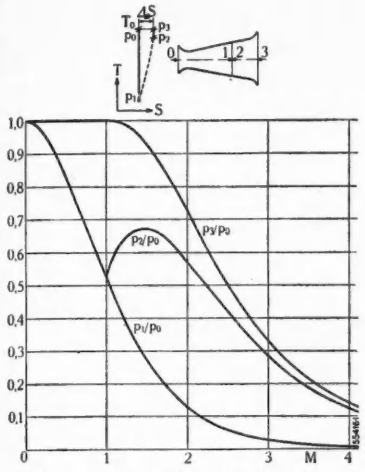


Fig. 5. Laws of the compression shock.

p_0. Pressure in front of the nozzle.
p_1. Pressure in front of the shock wave.
p_2. Pressure behind the shock wave.
p_3. Pressure after recuperation of the kinetic energy beyond the shock wave.

original temperature T_0 is restored by a frictionless recuperation of the velocity, the pressure p_3 will be lower than the initial value p_0 (Fig. 5).

This has important consequences. If after a compression shock the jet is once again accelerated, the sound velocity takes place in a section different from that before the shock, where the area is p_0/p_3 times as great. If now the second neck in our test section is not larger by at least this amount than the first one, a second sound velocity zone occurs there, whilst the first compression front still lies in the diverging part of the nozzle. It is not possible to get this compression front to pass over the point where it is swept down the test section (Fig. 6d).

The second neck must, therefore, be greater than the first one, at least in the ratio p_0/p_3, to which a margin should be added for the increasing thickness of the boundary layer and to insure stable operation.

Stable operation of the test section is, therefore, characterized as follows :—

Production of the supersonic velocity in the convergent-divergent nozzle, reduction of the velocity in the converging diffusor to a value slightly above that of sound, a small re-increase in the diverging diffusor, a sudden jump to subsonic velocity in the shock front and further reduction in the rear part of the diffusor. It is not surprising that the degree of recuperation of the kinetic energy decreases with increasing Mach number. At the moment when, during putting into service, the shock front reaches the test section, the second neck offers no further advantages as only subsonic speed exists therein. The compressor must give a pressure ratio just as if no converging-diverging diffusor were present, even if the load can be subsequently reduced. In spite of this a purely diverging diffusor would not be advantageous, because in order to keep the shock front away from the test section, it would be necessary to go to a velocity still higher than that in the test section, in part of this diffusor which would result in larger shock losses.

The difficulty of the pressure jump at the moment of starting up can be avoided if the supersonic speed is built up in a nozzle with adjustable elastic walls instead of in one with fixed walls, such as has been recently done by Professor Ackeret. Not only does such a

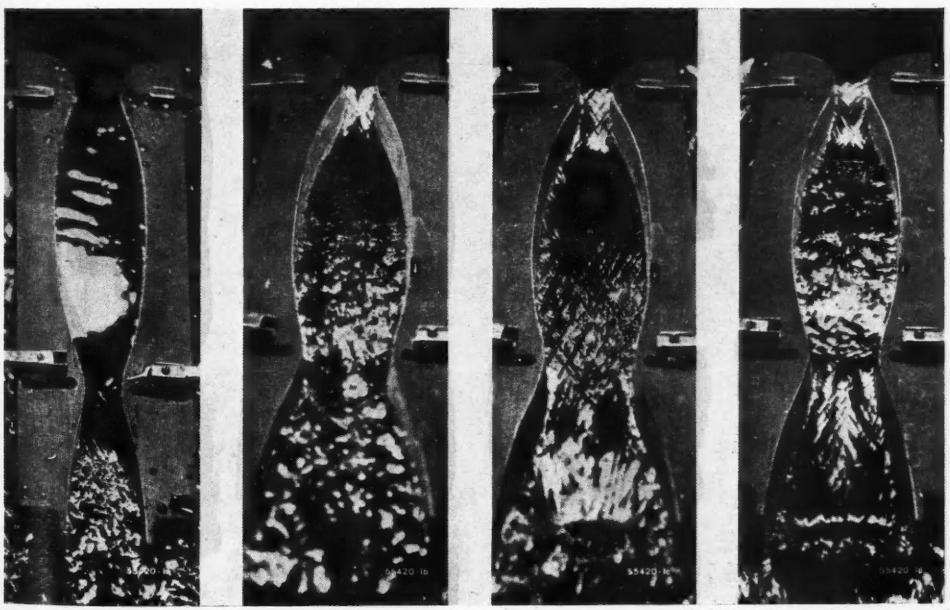


Fig. 6*a–d*. **Representation of the phenomena in the supersonic test by means of the flow of water in an open channel.**

(*a*) Appearance of the supersonic velocity behind the second, narrow neck.
(*b*) Appearance of the supersonic velocity behind the first neck during putting into service of a correctly dimensioned measuring section.
(*c*) Supersonic velocity in the measuring section. Compression shock behind the second neck (normal operation).
(*d*) Appearance of two pressure shocks, one after the first neck, the other after the second neck. Velocity in the test space below that of sound. The second neck is larger, but only slightly larger than the first one.

nozzle allow the Mach number to be varied continuously during the tests, but it also allows the ratio of the size of the second neck to the first one to be reduced and ensures a better recuperation of energy.

In order to achieve as nearly as possible parallel flow in the test section, the nozzle must be most carefully designed. The shape of the diverging part, in which the flow takes place above the velocity of sound, must in particular be given most careful attention. Every change of direction of the wall causes Mach waves to be set up. The waves, which originate in the neighbourhood of the neck of the nozzle travel across the channel and reach the opposite wall. It is necessary to ensure that at the point where they reach the opposite wall, a change of direction takes place, which sets up a wave of opposite sign and which, compounded with the first one, cancels it out. The waves originate at the convex wall and are destroyed at the concave one. The outline of a rectangular nozzle contained between two parallel walls can be theoretically determined by the Prandtl-Busemann method (Fig. 7). The method is due to Prandtl and Busemann, see Stodola Festschrift 1929, p. 499 or Prandtl: Abriss der Strömungslehre, p. 203. An analogous method has been developed by Darrieus for circular nozzles.

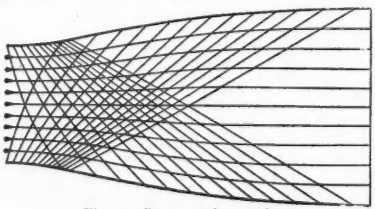


Fig. 7. Supersonic nozzle.
Determination of the outline of the divergent part of the nozzle for producing an eddy-free parallel flow in the test chamber by the Prandtl-Busemann method.

The equipment of the test section includes a precision barometer, measuring low absolute pressures. The suspension arrangement between the model and the balance is heavily loaded and must be located at a suitable place, for example, along the extension of the axes of a projectile model, in order that the waves shall not be disturbed. The suspension wires must be surrounded by fixed shrouds to protect them from wind pressures.

The conditions in the field of flow can be made visible by means of the Töpler method (Fig. 8). The field is illuminated, the light rays being refracted by the variations of pressure and throw a picture on the screen of the density gradients in the flow. In this manner it is possible to make visible Mach waves, shocks, separations, eddies, etc.

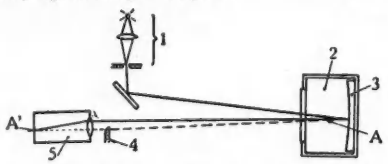


Fig. 8. Töpler process for making air flow visible.
1. Slit.
2. Measuring chamber in the wind tunnel.
3. Convex mirror.
4. Diaphragm.
5. Camera.

An upward rise of density at A deflects the light ray reflected by the convex mirror in an upward direction. The ray travels past the diaphragm 14, to the camera. The image A^1 of A appears light. A rise in density from bottom to top deflects the light ray in a downward direction. It is intersected by the diaphragm and A^1 appears dark.

THE BLOWER AND COOLER.

The pressure ratio to be delivered by the blower depends on the Mach number desired in the test section, the recuperation in the diffusor and the total resistance of the circuit. The following table is based on small scale nozzles, tested some by Professor Ackeret and some by Brown Boveri, and gives conservative figures which may be obtained with certainty.

Mach number ..	1	1.55	2	2.73
Pressure ratio of the nozzle	1.89	3.96	7.82	24.2
Pressure ratio of the diffusor	1.205	1.39	1.78	3.15
Pressure ratio of the compressor	1.265	1.46	1.87	3.3
Adiabatic compression power of the blower :				
Kinetic energy of the jet	0.42	0.35	0.45	0.68
Adiabatic compression power of the blower power without recuperation ..	0.35	0.24	0.25	0.27
Volume at suction of the blower for 1 m.² test section, m.³/sec.	251	237	220	202

The multi-stage axial blower used in the supersonic tunnel of the Swiss Federal Institute of Technology is shown in Fig. 9. Fig. 10 shows its characteristics.

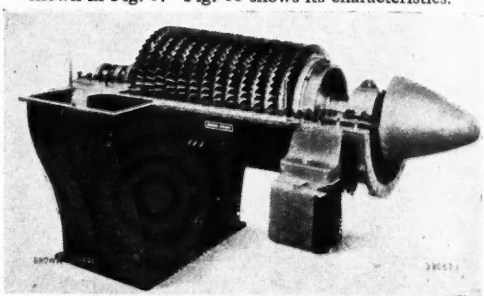


Fig. 9. Thirteen-stage axial blower for the Zurich wind tunnel.
The first multi-stage axial blower built by Brown Boveri at the conclusion of a long period of development.

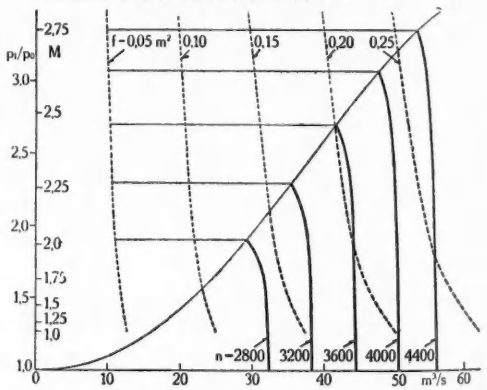


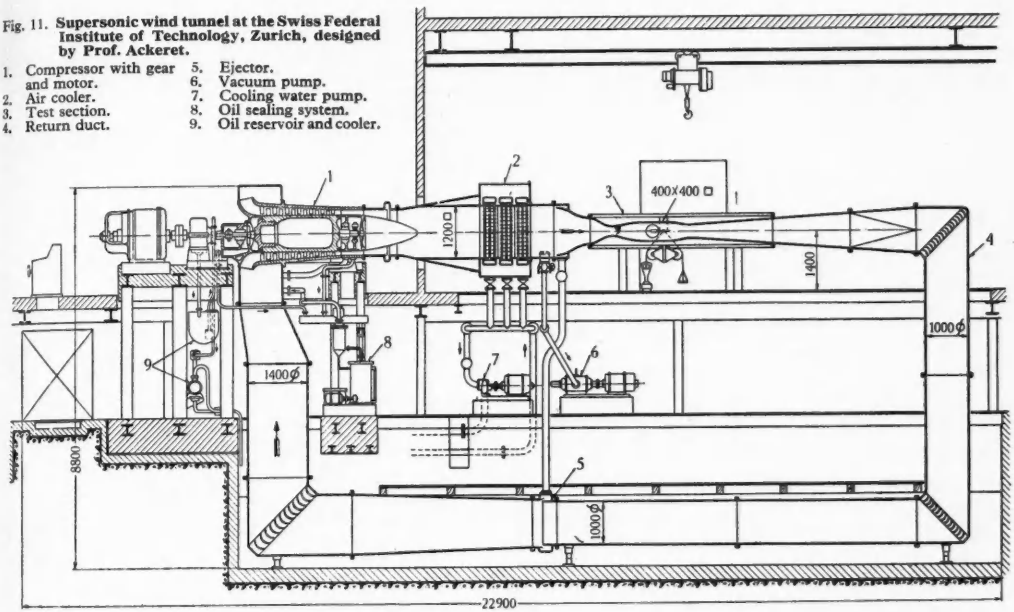
Fig. 10. Characteristics of a multi-stage axial compressor.
The diagram shows the Mach number which can be obtained and the corresponding sections of the jet *f*.

In a closed high-power wind tunnel the heat radiation from the walls of the channel is insufficient, and it is necessary to cool the air. It is convenient to use the cooler at the same time as a stabilizing honeycomb. For high Mach numbers the air is to be inter-cooled in the compression stages.

Fig. 11 shows the installation of the Zurich tunnel which might be looked upon as the prototype of modern design.

Fig. 11. **Supersonic wind tunnel at the Swiss Federal Institute of Technology, Zurich, designed by Prof. Ackeret.**

1. Compressor with gear and motor.
2. Air cooler.
3. Test section.
4. Return duct.
5. Ejector.
6. Vacuum pump.
7. Cooling water pump.
8. Oil sealing system.
9. Oil reservoir and cooler.



A.C. ELECTRICITY SERVICE METERS FOR 300 PER CENT OVERLOAD.

By M. WIESENGRUND. (From *Elektrotechnika*, Budapest, Vol. 37, No. 5, May 1st, 1944, pp. 120-128).

THE IMPORTANCE OF A NEW TYPE A.C. OVERLOAD METER FROM THE POINT OF VIEW OF ELECTRIC SUPPLY UNDERTAKINGS.

As long as a consumer only used a few electric lamps on his 220 V circuit, a standard 1.5 amp. meter was satisfactory and if more electrical household apparatus was to be connected up, a larger meter for 5.0 amps. was in general use.

These meters when new, start registering at about 0.5 to 1.0 per cent of their nominal load, but after a few years operation, due to an increase of frictional losses, the meters usually begin to register only at 2 to 2.5 per cent of their nominal load.

At very light loads all meters register to the disadvantage of the Supply Companies. These may be less than the actual consumption, and with a standard type of 5.0 amp. meter in use for a few years, it would be possible for a consumer to burn an electric lamp of 10 watts at no expense.

The new type of meter has been designed for an overload of 300 per cent ; the smaller type for 1.5 amp./ normal and 4.5 amp./max., the larger type for 3.0 amp./ normal and 9.0 amp./max. load at 220 V.

The torque of the smaller type at normal load of 330 watts is 5.0 gcm. and at max. load of 990 watts 15.0 gcm.

This meter, after several years of service will start registering at 2.5 per cent of the nominal load of 330 watts or at 8.3 watts, and will, therefore, register even on the load of a single 10 watt lamp.

Service meters begin to register correctly within the permissible limits of deviation at about 20 per cent of their nominal load.

The small new overload meter exerts a torque of 1.0 gcm. at 20 per cent of its nominal load of 330 watts, corresponding to 66 watts and will maintain its accuracy from this load upwards through many years of service.

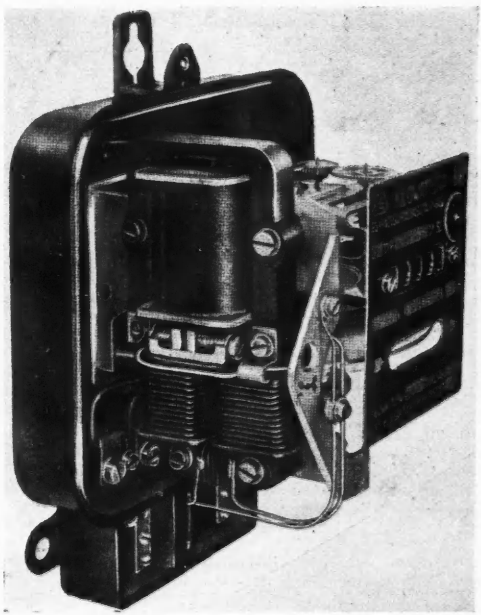


Fig. 1

The following table compares the data obtained from a standard 5.0 amp./220 V. meter with those of a new overload meter for 1.5 amp. normal and 4.5 amp. max.

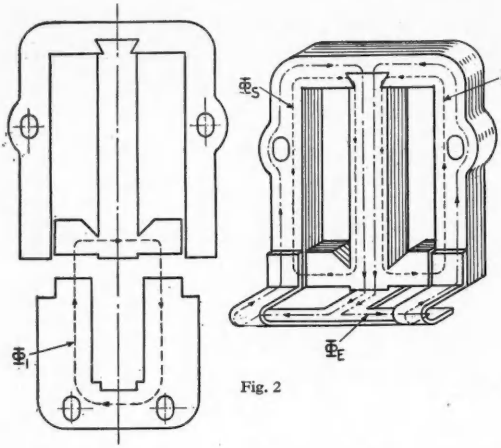


Fig. 2

	Standard type	New type
Normal load	1100 watts.	330-990 watts.
Starting registration when new, at 1 per cent	11. „	3.3 „
Starting registration when old, at 2.5 per cent	27 „	8.3 „
Torque of 1 gcm. at 20 per cent of normal load	220 „	66.0 „

With a view to reducing friction losses, the speed of the meter was fixed at 22 r.p.m. on normal load and at 66 r.p.m. on max. load.

With the introduction of overload meters, the Supply Companies need not keep such a large number of types of service meters in stock, as the two types of overload meter described above, can replace 5 types of standard meters ranging from 1.5 amp. to 15 amp. normal load.

In case of moderate extensions in household installations, it may not be necessary to exchange for a new type of meter, if its maximum permissible load will not be exceeded.

SHORT REVIEW OF THE THEORY AND OPERATION OF A.C. METERS.

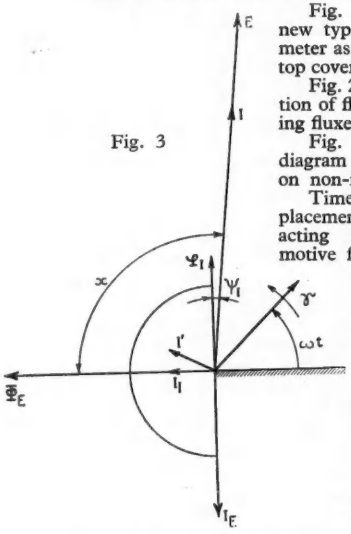


Fig. 3

Fig. 1 illustrates the new type "A" overload meter assembled and with top cover removed.

Fig. 2 shows the direction of flow of the operating fluxes.

Fig. 3 is a vectorial diagram of an a.c. meter on non-inductive load.

Time and phase displacements between the acting fluxes, electromotive force and current are shortly explained and necessary corrections achieved by means of an adjustable short-circuit coil applied to the current-magnet.

Fig. 4 shows a load and error curve of a standard a.c. meter without overload capacity.

The author analyses such load curves and describes his method of preventing the new type meter registering too slowly at light loads.

Fig. 5 illustrates the current-magnet with a magnetic-shunt over its two poles.

In spite of a certain drawback, introduced through the magnetic-shunt, the author succeeded in raising the torque of the new meter to 4.75 gcm. with a loss in the shunt coil of 0.64 watts.

The meter gave equally satisfactory results on inductive loads with a power factor of 0.50 and the percentage error was well within the permissible limits, specified by the Hungarian National Standard Institute.

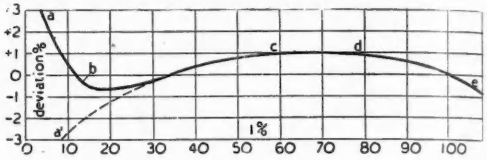


Fig. 4

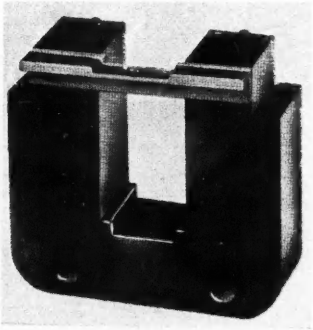


Fig. 5

CONSTRUCTIONAL FEATURES OF THE "A" TYPE OVERLOAD METER.

The author states his reasons for altering the size of the shunt coil wires, increasing their diameter to 0.13 mm. compared with the standard size of 0.10 mm. used generally in standard service meters.

The overload meters are provided with a very strong brake magnet, considerably more powerful than those used in standard meters. At the max. load a torque of 15 gcm. has to be considered, with a speed of 66 r.p.m. or 1 rev. per second.

Due to war time conditions cobalt magnets could not be obtained and an aluminium-nickel alloy, called "ergit" was used instead, and this was subjected to an ageing process.

The brake magnets are illustrated in Fig. 6. The two poles are arranged one on the top of the other, separated by an air gap of 2.3 mm. and they exert a very strong magnetic pull.

Temperature compensation is provided ensuring that the speed of the meters should not vary more than 0.3 per cent for each increase or decrease in temperature of 10 deg. C.

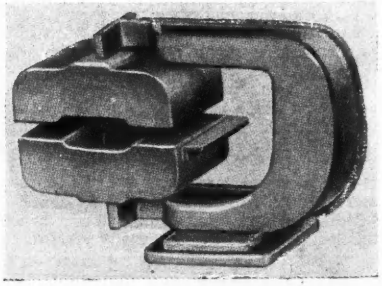


Fig. 6

NEW METHOD OF DETERMINING THE HEAT EXCHANGE THROUGH RADIATION.

By W. ORDINANZ. (From *Schweizer Archiv*, Vol. 10, No. 4, April, 1944, pp. 113-115).

EXCHANGE of heat between bodies of different temperature may take place by contact or by radiation[1]. While a number of simple formulae, charts and tables are at the disposal of the engineer for calculating the heat exchange through contact, no such means exist for determining the heat exchange through radiation.

Although the fundamental laws of radiation were clearly established by Boltzmann, Kirchoff, and others, it is not possible, except in special cases, to obtain satisfactory results, without having recourse to rather complicated mathematical formulae. This applies also to comparatively simple surfaces and even then, assumptions must be made to simplify calculations, as will be demonstrated below by means of an example of heat exchange through radiation of two bodies, situated at any arbitrary position in space.

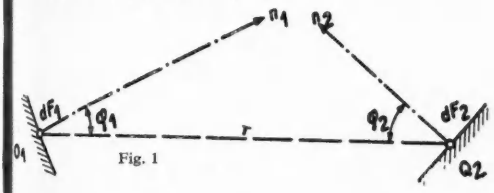


Fig. 1

The following symbols are used in Fig. 1 :—

T_1, T_2 Absolute temperatures in deg. K. of the two bodies I and II, C_1, C_2 Radiation coefficients of the two bodies in kcal./m².*hr.* (deg. K.)[4], C_s Radiation coefficient of an ideal Blackbody = 4.95, dF_1, dF_2 Surface elements of the two bodies at points O_1, and O_2. φ_1, φ_2 Angles formed by the lines O_1, O_2 and the perpendiculars to the surface elements dF_1, and dF_2.

r Distance between O_1, O_2.

$dQ_{1\to2}$, $dQ_{2\to1}$ Quantity of heat radiated from dF_1 to dF_2 and from dF_2 to dF_1

$dQ_{1\text{-}2}$ Quantity of heat passing between dF_1 and dF_2 through radiation, equal to the arithmetical sum of $dQ_{1\to2}$ and $dQ_{2\to1}$.

According to Stefan-Boltzmann, Lambert, etc., the following equations can be established :

$$dQ_{1\to2} = \frac{C_1}{\pi}\left(\frac{T_1}{100}\right)^4 \frac{\cos\varphi_1 \times \cos\varphi_2}{r^2} \times dF_1 \,.\, dF_2$$

In the case of actual bodies, however, only a part of this heat is absorbed by dF_2, while the balance is reflected into space of which again only a part falls on dF_1; a part of it is absorbed, the rest reflected and so forth.

In the opposite direction, the surface element dF_1 receives the following amount of heat from dF_2,

$$dQ_{2\to1} = \frac{C_2}{\pi}\left(\frac{T_2}{100}\right)^4 \frac{\cos\varphi_1 \times \cos\varphi_2}{r^2}\, dF_1 \,.\, dF_2$$

Similarly only part of this heat is absorbed by dF_1 and the remainder is reflected into space, of which again only a part reaches dF_2, etc.

To determine the exact values of only a few links in this to- and fro-radiation, and to establish a heat balance of the heat exchange between the two surface elements dF_1 and dF_2, some assumptions must be made.

For bodies with such a high absorption capacity, that the radiated heat may be considered almost completely absorbed on the first impact, the heat exchange between dF_1 and dF_2 can be calculated with sufficient accuracy according to Nusselt's[2] formula :

$$dQ_{1\text{-}2} = \frac{C_1}{\pi}\,\frac{C_2}{C_s}\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \cdot \frac{\cos\varphi_1 \times \cos\varphi_2}{r^2} dF_1 \,.\, dF_2$$

and the total quantity of heat exchanged through radiation for the whole of the surfaces becomes :

$$Q_{1\text{-}2} = \frac{C_1}{\pi}\,\frac{C_2}{C_s}\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \cdot \int_{F_1}\int_{F_2} \frac{\cos\varphi_1 \times \cos\varphi_2}{r^2} dF_1 \,.\, dF_2$$

In spite of simplifications to be introduced, the solution of the double integral, known as "Angle coefficient" or "Angle-ratio" meets with some difficulties, even in the case of simple geometrical configurations.

In practice one has recourse to different graphical methods of approximation, such as proposed by Nusselt,[3] Hottel[4] and Seibert[5]. For cases most frequently met with, computed values of *Angle coefficients*[6] can be found in the literature.

An entirely new method, distinguished by its great simplicity, has been published under the name *Flux Algebra*[7], by a Russian Author, Pollak[8] and a short introduction of this purely analytical method should be of interest to the professional world, the more so, as this method has so far only been mentioned in the original publication.

To start with, the two fundamental conceptions, with which the *Flux Algebra* is operating need to be explained, namely the *Angle coefficient* and the *Effective surface.*

The *Angle coefficient* or *Angle relation* of two bodies also called by Pollak the *Radiated coefficient* is determined by the ratio of the proportion of radiation falling on one of the bodies, to the total radiation emitted by the radiating body.

Let Q_1 be the total radiation emitted by body I and $Q_{1\to2}$ be the portion of it falling on body II., the *Angle coefficient* will be :

$$\varphi_{1\text{-}2} = \frac{Q_{1\to2}}{Q_1}$$

Effective surface of body I in relation to body II means the numerically equivalent part of the radiating surface of body I corresponding to the *Angle coefficient.*

If $E_{s\text{-}1}$ (in kcal./m². hr.) is equal to the black radiation of a unit surface of body I, then :

$$Q_1 = E_{s\text{-}1} \,.\, F_1$$

Similarly one can express $Q_{1\to2} = F_{s\text{-}1} \,.\, H_{1\text{-}2}$ as a product of the specific radiation multiplied by a theoretical surface, bearing the same relation to F_1, as $Q_{1\to2}$ to Q_1

$$Q_{1\to2} = \varphi_{1\text{-}2} \,.\, Q_1 \text{ and } H_{1\text{-}2} = \varphi_{1\text{-}2} \,.\, F_1$$

The effective surface is also called by Pollak the mutual radiation surface.

The radiation of body I to body II and body III is the same, the angle of space and their position being the same (Fig. 2).

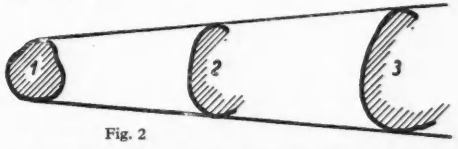


Fig. 2

The conceptions of angle-coefficient and mutual radiation surfaces are bound up with the existence of a pair of bodies. The numerical values of both are the

1, 2, etc. See Bibliography at the end of the article.

same, but the angle coefficient has no dimensions, whereas the mutual radiation surface represents an area.

With regard to the manner of notation the author proposes to use indices with an arrow for the radiated heat quantities which have in certain aspects a vectorial character. As an example, $Q_{1\to 2}$ is contrary to the radiation from the opposite direction, $Q_{2\to 1}$, and the resultant radiation Q_{1-2} is denoted without an arrow in the index. For H and φ there is no need to use arrows in the index.

The author is basing his *Flux Algebra* on the following five laws of radiation :

(1) The radiation from a body to several different bodies lying behind one another is the same irrespective of the distances, provided that the space angle and position is the same.

The mathematical formula expressing this law is as follows :

$$Q_{1\text{-}2}=\varphi_{1\text{-}2}\,.\,F_1\,.\,C_o\left(\frac{T_1}{100}\right)^4=Q_{1\text{-}3}=\varphi_{1\text{-}3}\,.\,F_1\,.\,C_o\left(\frac{T_1}{100}\right)^4$$

and consequently :

$$\varphi_{1\text{-}2}=\varphi_{1\text{-}3}\,;\ H_{1\text{-}2}=H_{1\text{-}3}$$

(2) The total heat exchange between bodies is equal to the arithmetic sum of radiation between the individual surface elements.

If, for instance, body 1 is divided into the surface elements 1a and 1b and the body 2 into 2a and 2b, the following equation is valid :

$$Q_{1\to 2}=Q_{1a\to 2a}+Q_{1a\to 2a}+Q_{1b\text{-}2b}$$

Consequently :

$$H_{1\text{-}2}=H_{1a\text{-}2a}+H_{1a\text{-}2b}+H_{1b\text{-}2a}+H_{1b\text{-}2b}$$

(3) If an opaque screen is placed between two bodies, the exchange of radiation becomes nil and $Q_{1\text{-}2}=O$.

It follows :

$$H_{1\text{-}2}=O \text{ and } \varphi_{1\text{-}2}=O.$$

For bodies without concave surfaces :

$$Q_{1\text{-}1}=O,\ H_{1\text{-}1}=O,\ \varphi_{1\text{-}1}=O.$$

With bodies having concave surfaces self-radiation may occur.

(4) If a radiating body 1 is surrounded by $(m\text{-}1)$ bodies, the following equation may be used :

$$\sum_{i=1}^{i=m} Q_{1\to i}=F_1\,.\,C_o\,.\left(\frac{T_1}{100}\right)^4 \text{ and similarly :}$$

$$\sum_{i=1}^{i=m} H_{1\text{-}i}=F_1 \text{ and } \sum_{i=1}^{i=m}\varphi_{1\text{-}i}=1.$$

For a geometrically closed system of n bodies, one can establish n equations, as follows :

$$\sum_{i=1}^{i=n} H_{k\text{-}i}=F_k \text{ where } k=1, 2, 3 \ldots\ldots n. \text{ and}$$

$$\sum_{i=1}^{i=n}\varphi_{k\text{-}i}=1 \text{ where } k=1, 2, 3 \ldots\ldots n.$$

(5) For Blackbodies of identical temperatures, the mutual radiations are equal, independently of their position in space and of the medium between them,

$$Q_{1\to 2}=Q_{2\to 1}$$

Since the temperatures and the radiation coefficients are equal, it follows that :

$$H_{1\text{-}2}=H_{2\text{-}1} \text{ or } F_1\,.\,\varphi_{1\text{-}2}=F_2\,\varphi_{2\text{-}1}$$

The following example should demonstrate how easy it is to carry out the calculation by using the rules of the *Flux Algebra*.

The heat exchange through radiation is to be determined between a row of boiler water tubes and a parallel wall of the hearth (Fig. 3).

According to rule 1 of the *Flux Algebra* one might substitute for the hearth wall another one tangential to

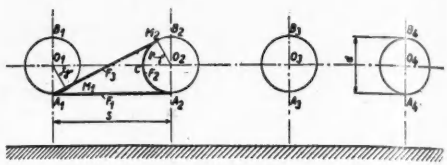


Fig. 3

the tubes. It follows from rule 3 that the radiation from one tube can reach only the part $A_1\,A_3=2\,A_1\,A_2$ of the wall surface.

For the geometrically closed system of surfaces :

$$A_1\,A_2=F_1,\ A_2\,M_2=F_2,\ A_1\,M_1+M_1\,M_2=F_3$$

According to rule 4 one can establish the following equation :

$$\left.\begin{aligned} H_{1\text{-}1}+H_{1\text{-}2}+H_{1\text{-}3}&=F_1\\ H_{2\text{-}1}+H_{2\text{-}2}+H_{2\text{-}3}&=F_2\\ H_{3\text{-}1}+H_{3\text{-}2}+H_{3\text{-}3}&=F_3\end{aligned}\right\}\ \ldots\ldots\ (a)$$

where F_3 may be considered as an imaginary auxiliary surface (parameter). As none of these surfaces are concave, rule 3 can be applied :

$$H_{1\text{-}1}=O,\ H_{2\text{-}2}=O,\ H_{3\text{-}3}=O.$$

Assuming identical temperatures and ideal Blackbodies (any deviations from this may be corrected later by applying approximate coefficients), it follows from rule 5 that :

$$H_{1\text{-}2}=H_{2\text{-}1}\,;\ H_{1\text{-}3}=H_{3\text{-}1}\,;\ H_{2\text{-}3}=H_{3\text{-}2}$$

Substituting the above relations in (a), adding the three equations and subtracting from the sum the equations for F_1, F_2, F_3, we obtain :

$$\begin{aligned} H_{1\text{-}2}&=\tfrac{1}{2}\,(F_1+F_2-F_3)\\ H_{2\text{-}3}&=\tfrac{1}{2}\,(F_2+F_3-F_1)\\ H_{1\text{-}3}&=\tfrac{1}{2}\,(F_1+F_3-F_2)\end{aligned}$$

The mutual radiation surface between the tubes and the wall is :

$$H_{w\text{-}r}=H_{r\text{-}w}=2\,H_{1\text{-}2}=F_1+F_2-F_3$$

The radiation coefficients from tube to wall :

$$\varphi_{r\text{-}w}=\frac{H_{1\text{-}2}}{F_r}=\frac{F_1+F_2-F_3}{F_r},$$

from wall to tube :

$$\varphi_{w\text{-}r}=\frac{H_{1\text{-}2}}{F_\psi}=\frac{F_1+F_2-F_3}{F_1}$$

Let s be the pitch of the tube distances, d the diameter of the tubes, and γ the angle at point of tangent :

$$\overline{A_1A_2}=s\,;\ \overset{\frown}{A_2CM_2}=(\pi/2+\gamma)\,\frac{d}{2}\,;\ \overset{\frown}{A_1M_1F_3M_2}=\overset{\frown}{A_1M_1}+$$

$$\overline{2\,M_1F_3}=(\pi/2-\gamma)\,\frac{d}{2}+d.\,tang\,\gamma.\quad \text{Thus}$$

$$H_{r\text{-}w}=H_{w\text{-}r}=s-(tang\,\gamma-\gamma)\,.\,d$$

From the triangle (Fig. 3) $O_1M_1F_3$:

$$\gamma=tan^{-1}\sqrt{(s/d)^2-1}$$

$$H_{r\text{-}w}=H_{w\text{-}r}=s-d\sqrt{(s/d)^{2-1}}+d\ tan.\ \sqrt{(s/d)^{2-1}}$$

For quick calculations one may apply an approximation, called the *Thread rule.*

If one imagines an elastic thread laced around and completely fitting the contours of two bodies which are subjected to heat exchange (Fig. 4), the mutual radiation surface is equal, according to Pollak, to one-half of the length of the interior threads less one-half of the length of the exterior threads.

$$H_{1\text{-}2}=H_{2\text{-}1}=\frac{A_1C_1D_2B_2+B_1D_1C_2A_2}{2}-\frac{A_1A_2+B_1B_2}{2}$$

Based on this rule, one can write down at once the result of the calculation, which was obtained for the previous example.

The original publication, as mentioned, gives a list of computed angle-coefficients for 43 different instances, a number of which might, of course, also be determined by different though complicated methods.

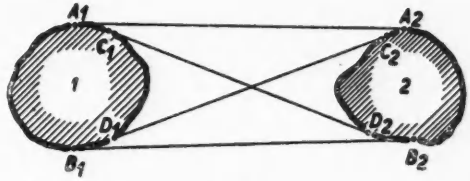


Fig. 4

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DYNAMIC SIMILITUDE OF TURBO-MACHINERY FOR COMPRESSIBLE FLUIDS.

By M. SÉDILLE. (From *La Technique Moderne*, Vol. 35, Nos. 23/24, December 1st and 15th, 1943, pp. 175-182).

THE principles of similitude for incompressible flow are generally applied with advantage to turbo-machines, such as pumps and hydraulic turbines. They are also applicable to fans, in which case the working fluid can be considered as practically incompressible.

In the case of steam or gas turbines, matters are, however, complicated by the compressibility of the fluid, so that Rateau's similarity concepts are no longer applicable. But, by taking into account the compressibility of the working fluid, similarity laws can be established which represent a generalisation of the corresponding laws derived for incompressible fluids and are, as such, contained in the original concepts established by Rateau with regard to incompressible fluids.

According to the principle of similitude laid down by Combes-Rateau, similarity is defined by the characteristic dimensionless coefficients $\mu = \frac{gH}{u^2}$ $\delta = \frac{Q}{u\,r^2}$ ρ

where H is the increase in the mechanical energy of the fluid per unit of weight in the case of compressors, Q is the volume delivery of the machine, ρ is its internal efficiency, and u and r respectively are a typical velocity and dimension. The two latter are usually represented by the peripheral velocity and the wheel radius. Analogously, the power coefficient is established as $\tau = \frac{gW}{\bar{\omega}\,u^3\,r^2}$

where W is the input and useful power at the shaft of the machine and ω is the density of the fluid. For power generating machines (turbines) it is $\tau = \mu\,\delta/\rho$, while for power absorbing machines (pumps) it is $\tau = \mu\,\delta\rho$. Based on laws of similitude, a graph can be constructed on the basis of the functionality $\mu\ \rho = f_{1,2}\,(\delta)$. A graph of this kind shows the operating characteristics at different speeds with whatever incompressible fluid is used, not only of the machine considered, but of all machines which are similar. But this statement must be qualified, insofar as the differences in speeds and dimensions must not be unreasonably great, as this would introduce variations in efficiency owing to the influence of the Reynolds number and the relative surface roughness.

Generally, the functional relationship between the dimensionless coefficients can therefore be written $\mu\ \rho = f_{1,2}\,(\delta_1\ Re_1\ h/r)$, where $Re = ur/\nu$ is the Reynolds number, ν is the kinematic viscosity of the fluid, and h/r represents the relative roughness. The factor h may be defined, for instance, by a mean dimension of the wall roughness. These relations are established on the assumption that nowhere in the path of flow is cavitation present. In cases where cavitation occurs, the condition of similitude will be met by a constant value of $(p-e)/\rho u^2$ where e is the vapour tension prevailing under operating conditions.

In the case of elastic fluids in which thermal exchanges are zero or negligible, two further parameters must be employed. Taking compressibility into account, these are the coefficient γ, representing the ratio of specific heats and the Mach number relating a reference velocity to that of sound at a given point of flow.

With regard to turbo-machines, the compressibility likewise requires the introduction of two additional parameters in order to satisfy conditions of similarity. A further complication is engendered by the fact that the definition of the coefficients μ, δ, and ρ can only be applied with reservations to elastic elastic fluids.

The various independent variables are P_o, T_o (or a_o and $\bar{\omega}_o$, the velocity of sound and the density of the fluid at the point of admission to the machine), γ, r, u, and the rate of discharge by weight I. According to the laws of similitude, these six variables can be reduced to three dimensionless variables (Theorem of Vaschy). Thus, by neglecting the influence of the Reynolds number and of the relative roughness, the characteristic relation can be expressed in the form

$$\frac{P_1}{P_o}\ \rho = f_{1,3}\left(\frac{\mu}{a_o}\ \frac{I}{a_o\,r^2\,\bar{\omega}_o}\ \gamma\right)$$

Here the variable $\frac{\mu}{a_o}$ represents the Mach or Bairstow number. This is ratio of a reference velocity (in this case the peripheral speed of the wheel) to the velocity of sound at a reference point, for the sake of convenience the inlet being chosen as reference point. The member $I/a_o\,r^2\,\bar{\omega}_o$ which can be designated as the reduced delivery, replaces the characteristic delivery factor $Q_o/a_o\,r^2$ where a_o represents the suction volume. Finally the compression ratio has taken the place of the manometric coefficient.

Figs. 1 and 2 represent the general relations for an aircraft engine supercharger and an axial-flow centrifugal blower respectively for cases where a change in the nature of the fluid does not occur, or where the respective fluids have the same specific weight. The efficiency values shown, represent the polytropic efficiency which has been chosen instead of adiabatic efficiency, since it approaches more closely to the real efficiency. These graphs possess the advantage of enabling an immediate determination of the influence upon performance exerted by a variation in intake conditions, as a variation in altitude in the case of the aircraft engine supercharger. Performance curves of this kind can be established without regard to the temperature or intake pressures prevailing and make it possible to establish by experiment alone, whether specified operating conditions will be obtained at the altitude in question. Experi-

mental proof of the exactness of these laws has been given by Capon and Brooke*. A graphic representation of this type is independent of the nature of the fluid employed provided that gases of same atom number are involved. Should this not be the case, the coefficient γ, that is, the ratio of the specific heats, may have a certain influence and thereby modify the shape of the curves.

The form of the dimensionless coefficients given above, is arbitrary and was chosen for the sake of convenience. Actually, the form of these coefficients is only one among many possible forms. Thus, in a recent investigation†, Kuehl has—in accordance with German practice—replaced the delivery coefficient $I/a_0 r^2 \omega_0$ or $Q_0/a_0 r^2$ by the ratio of the velocity at the entrance section to the peripheral velocity, writing $\varphi = \frac{v_0}{\mu}$. For certain purposes this formulation may be more suitable than any other. This holds true, for instance, in the case of aircraft engine superchargers, where it is especially instructive to plot the performance characteristics with $\bar{\omega}_1/\bar{\omega}_0$ as ordinate and $I/a_0 r^2 \bar{\omega}$, or $Q_1/a_0 r^2$ as abscissa. Here Q_1 designates the delivery effected for different values of the Mach number u/a_0. A graph of this kind may also include the curves of equal power absorption. A typical example is given in Fig. 3, where for given admission conditions (a_0 and ω_0) depending on the altitude and the speed of the aircraft, the graph shows directly the specific amount of air admitted to the cylinders and also, the corresponding power consumption of the supercharger.

An important point to be mentioned is, that in order to satisfy conditions of similitude, the various clearances of the machine and especially the clearances between moving and fixed parts must be kept in constant ratio to the dimensions of the machine. In certain cases, these clearances will have an important influence upon performance characteristics. The clearances will, however, be affected by elastic and thermal deformations of both impeller and casing, the exact extent of the de-

*Reports and Memoranda, No. 1336.

†Forschung, Nov./Dec., 1942.

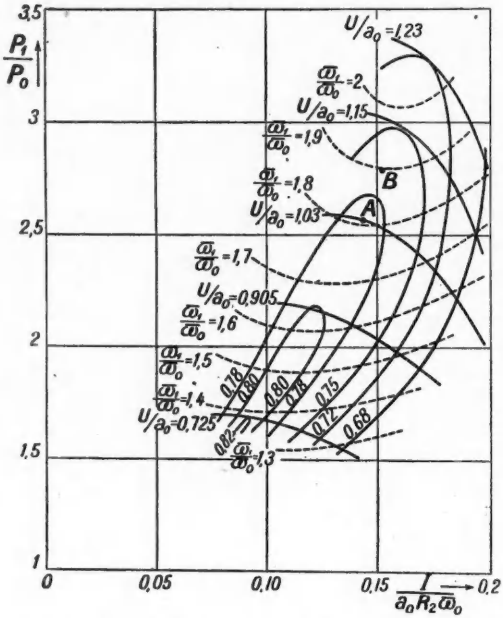


Fig. 1. **Characteristics of an aircraft engine supercharger.**

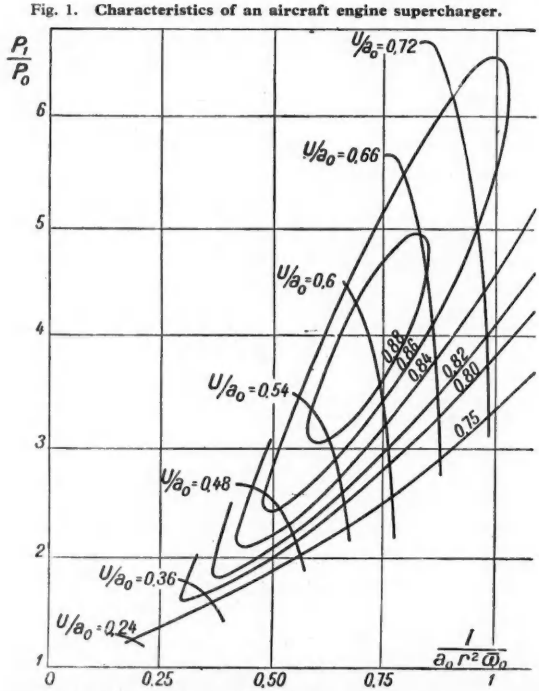


Fig. 2. **Characteristics of an axial-flow compressor.**

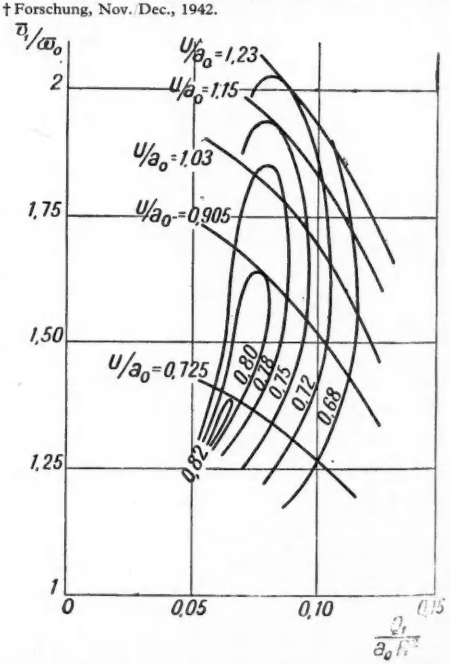


Fig. 3. **Characteristics of an aircraft engine supercharger established from the aspects of the adaptability of the supercharger to the aircraft engine.**

formations depending upon the mechanical properties of the material.

In the case of aircraft engine supercharger, for instance, the actual barometric pressure prevailing at the intake will be given by the atmospheric conditions. When testing the supercharger on the test stand, overloading of the impeller under normal atmospheric conditions must be avoided by throttling the intake at max. speed; while in actual operation this maximum speed is only employed at the corresponding altitude. Throttling the air supply in test stand trials may cause a certain flattening of the casing, which leads to a decrease of lateral clearances, but in actual operation, the decreased atmospheric pressure will have exactly the opposite effect—that is, the clearances will tend to open up. Exact similarity will not, therefore, prevail in the two cases considered. Similar considerations apply to the loss of heat from the casing to ambient atmosphere; and the transfer of heat from the admission to the delivery side, through the metal of the casing, may likewise affect performance characteristics to a certain extent. Such heat transfer may be small enough to be neglected as far as aircraft engine superchargers are concerned. But according to test data obtained by the Société Rateau, these heat losses may exceed as much as five per cent of the output of industrial compressors. This matter becomes still more serious where the compression of a hot fluid is concerned.

Taking the case of compressors equipped with one cooling circuit, the requirements of similitude will be met if the amount of heat transferred to the cooling water comprises a constant fraction of the energy expended upon compression. With an overall heat transfer coefficient k, the amount of heat absorbed by the cooling water will be given by $k\,r^2\,(T_1 - T_{11q.})$ where $T_{11q.}$ is a reference temperature of the cooling circuit. As the delivery of compressed fluid is proportionate with $\bar{\omega}\,ur^2$, the amount of heat extracted per unit weight of the fluid will be given by $\dfrac{k\,(T_1 - T_{11q.})}{\rho\,g\,u}$. Therefore if α denotes the ratio of energy absorbed by the cooling liquid to the energy expended upon compression, this coefficient will vary with $\dfrac{k\,(T_1 - T_{11q.})}{(\rho\,g\,c_p)\,u\,(T_1 - T_o)}$. By introducing the Nusselt function $Nu = \dfrac{k\,r}{\lambda_{gas}}$ we have

$$\frac{Nu\,S_{gas}\,(T_1 - T_{11q.})}{Re_{gas}\,(T_1 - T_o)}.$$

Like Nu, α also depends upon the variables which affect the coefficient of heat transfer from the gas to the water. Or in other words, the coefficient will vary with the Reynolds number and with the Stanton number, as well as with the thermal conductivity of gas and liquid. Finally, it will also vary with the thermal conductivity of the wall unless this should prove negligible in the case of heat flow involved. Thus it is seen that in addition to the mechanical factors, the following dimensionless coefficients must be taken into account, to wit:—

$$\lambda_{11q.}/\lambda_{gas}/\lambda_{metal},\ Re_{11q.},\ Re_{gas},\ S_{gas},\ \frac{I_{11q.}}{I_{gas}},\ \frac{T_{11q.}}{T_{gas}},\ \frac{C_{p\,11q.}}{C_{p\,gas}}$$

If factors of secondary importance such as the influence of the Reynolds number upon the mechanical characteristics, are neglected, and if no change takes place in specific heat and in the conductivities involved, the most important thermal factors will be $I_{11q.}/I_{gas}$ and $T_{11q.}/T_{gas}$. Provided a given case admits of this simplification, the characteristics of a compressor with cooling circuit will be given by

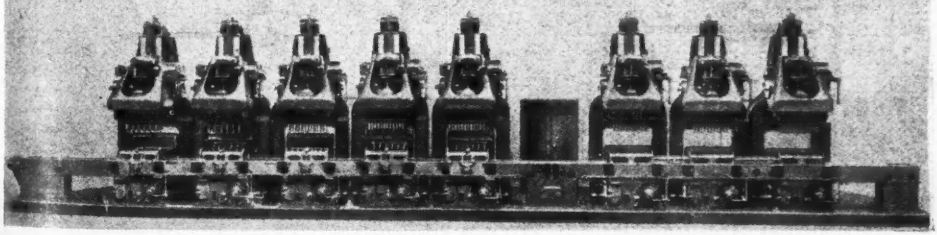
$$\frac{P_1}{P_o} \quad \rho = f_{1,2}\left\{ \frac{u}{\alpha_o} \quad \frac{I}{\alpha_o\,r^2\,\bar{\omega}_o} \cdot \frac{T_{11q.}}{T_{gas}} \quad \frac{I_{11q.}}{\alpha_o\,r^2\,\bar{\omega}_o} \right\}$$

Similar considerations apply to compressors equipped with inter-coolers.

AUTOMATIC TRANSFER MACHINES FOR DRILLING CYLINDER BLOCKS IN MASS PRODUCTION

(From *Die Werkzeugmaschine*, Vol. 47, No. 6, March, 1943, p. 132).

THIS automatic transfer machine serves for the simultaneous drilling, countersinking, and tapping of all bolt holes of a cylinder block. An automatic transfer mechanism with pneumatically operated clamping devices conveys the work pieces to the individual stations, the drives of which are started by push button control. The tool heads then move forward in rapid advance and the holes are drilled at the normal rate of feed, and the tool heads returned automatically to their home position. For tapping operations, special reversing motor drives are provided. For this purpose each spindle is equipped with an electric limit switch which is closed as the multi-spindle drill head returns into its home position. The work piece, placed on a carrier plate, is automatically moved on the hardened ways from station to station. There are eight stations with a 180 deg. turn after the fifth station, a finished piece leaving the line every three minutes. The total machining time therefore amounts to some 27 minutes. The longest drilling time is that for the tappet holes, which takes some two minutes. To this must be added the time for automatic clamping, unclamping, and moving the carrier plate to the next station, which takes about one minute. Thus there are two minutes available at each station for carrying out the machining operation, but this time may not be entirely taken up at every station. Each drilling and tapping spindle set comes to rest independently after the termination of its operation, and only the termination of the last operation releases the conveying mechanism.



MICROCRYOSCOPIC DETERMINATION OF MOLECULAR WEIGHT

By Prof. Dr. G. Jung. (From *Chemische Technik*, Berlin, Vol. 16, No. 19, September 25th, 1943, pp. 189–190.)

Cryoscopic methods for determination of the molecular weight may be classified in two groups. In the first solvents with large molecular freezing point depressions are employed; while in the second method the considerable thermal capacity of the Beckmann thermometer is eliminated by using thermocouples or resistance thermometers. Although the latter methods can also be employed in the case of solvents with small molecular freezing point depressions, the use of thermocouples and of sensitive galvonometers proves rather cumbersome. Therefore, a new method has been developed which is applicable to any kind of solvent, regardless of its freezing point and its cryoscopic constant. As many liquids tend to undercooling because of the slow cooling rate required, observation of the appearance of the first crystals in the solvent cannot be resorted to. The melting point is, therefore, determined as the point at which the last crystals disappear from a solution during the melting process. Care must however be taken to ensure effective mixing of melting and existing liquid solution. This is achieved by making the melting process sufficiently slow and by using very narrow capillaries.

The apparatus used is shown in Figs. 1 and 2. It consists mainly of a copper body of 5 × 5 × 15 cm. with a 30 mm. bore for receiving the Beckmann thermometer and the heating bath, the necessary connections being made by means of ground joints being provided for this purpose. According to the temperature range required, water or glycerine-water baths are used. Stirring of the bath is done by air admitted through a small tube. Heating or cooling of the copper body is effected by passing hot water or brine through a number of inter-connecting passages drilled in the block. Circulation of heating or cooling liquid is carried out by means of a Höppler thermostat. During the test the temperature of the copper body is slowly increased so that the temperature of the bath increases by 0.05—0.07 deg. C. per minute. To increase the sensitivity of temperature control, electric heating of the copper body is also provided for. This is particularly useful when working at temperatures exceeding some 50 deg. C.

For illumination and observation of the capillaries, openings (F) are provided in opposite sides of the copper body, the observation opening being covered by a photographic plate with its sensitized layer (fixed without previous exposure) next to the copper body in order to prevent condensation of moisture on the glass pane. The illumination window is covered with a piece of frosting glass. Observation of the capillaries is carried out either with a Zeiss-Winkel microscope (10 × magnification) or with a magnifying glass. Particularly advantageous is the employment of polarised light, the arrangement used being shown in Fig. 3.

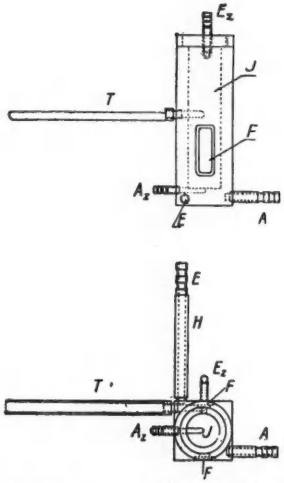
Capillaries of 0.02–0.04 mm. diameter and 12/13 mm. length are used. They are affixed horizontally to the bulb of the Beckmann thermometer by means of a special holding device. The horizontal position is resorted to in order to prevent the crystals formed from accumulating in the lower part of the capillary, with an attendant difference in concentration. Generally the bath in which the capillaries are submerged is heated solely by conduction through the air space between copper body and bath. In the investigation of solvents with elevated fusion points, as, for instance, of naphthalene, the air space between copper body and bath may be filled with a heat conducting liquid so as to shorten the testing time. Upon approach of the bath temperature within a few degrees to that to be measured, the heat conducting liquid is drained off in order to decrease the rate of temperature rise.

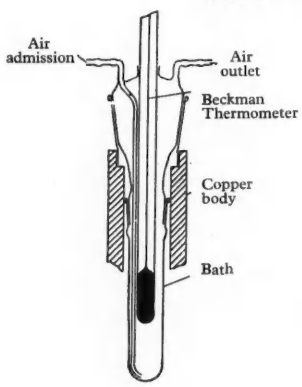
The accuracy obtained with this device may be judged from the data given hereunder:

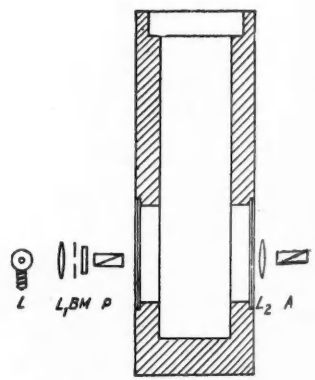
Solvent: Benzol.
(*Molec. freezing point depression* 5.8 *deg. C.*).
0.4015 g. Azobenzol in 10.8444 g. benzol.
Freezing point depression 1.036 deg. C.
Molecular weight found as 181.2 as compared to a computed value of 182.1.

Solvent: Glacial Acetic Acid.
(*Molec. freezing point depression* 16.65 *deg. C.*).
0.0415 g. Azobenzol in 1.0491 g. glacial acetic acid.
Freezing point depression 0.85 deg. C.
Molecular weight found as 181.5 as compared to a computed value of 182.1.

Solvent: Phenol.
(*Molecular freezing point depression* 41 *deg. C.*).
0.01085 g. Phenacetin in 0.5692 g. Phenol.
Freezing point depression 0.777 deg. C.
Molecular weight found as 59.8 as compared to a computed value of 60.05.







Figs. 1 and 2. **Apparatus for the determination of Freezing Point Depression.**
J Bore in copper body
E Admission of heating liquid
H Electrical heating equipment
A Exit of heating liquid
F Illumination and observation window resp.
T Thermometer sleeve
E_z Inlet } of conducting liquid
A_z Outlet }

Fig. 3. **Optical equipment.**
L Source of light
L_1 Lens
B Diaphragm aperture
M Frosting glass plate
P Polarizer
L_2 Magnifying glass
A Analyzer

HOW TO IMPROVE ACCURACY AND EFFICIENCY IN MEASUREMENT AND TESTING OF GEARS.

By Dr.-Ing. K. BÜRGER VDI, National Institute of Technical Physics, Berlin). (From *Maschinenbau der Betrieb*, Vol. 22, No. 3, March, 1943, pp. 101–107.)

WHILST a high degree of accuracy in measuring gear and toothed-wheels is possible by modern methods, it is not always simple to carry out, because of the complicated geometrical form of the work-piece as well as the construction of the machine tool necessary for production. Mutual agreement between user and producer should, therefore, always be obtained beforehand as to whether the degree of accuracy demanded by the former is really required for his purpose, and whether such requirements can be fulfilled within economic limits.

The main requirements in gears generally are :—

1. As far as possible they should run noiselessly and without vibration.
2. The efficiency should be high, i.e., low heating during the run ; copious lubrication is, therefore, of great importance.
3. High tensile, bending and fatigue strength of the teeth as well as high impact resistance.
4. Small wear and tear of the flanks and, in consequence, a long life of the toothed-wheels.
5. Transmission of movement should take place with a high degree of accuracy, especially with precision gears, e.g., machine tools, measuring and control apparatus.

One or other of these requirements may be of special importance according to the purpose for which the gear is to be used, but usually, maximum values are asked for simultaneously for all five points. It is clear that collectively they can only be the result of a very high degree of accuracy in production with tolerances of few μ as well as adequate precision measuring and controlling equipment. Moreover, highly skilled labour is necessary for working the complicated and expensive machines and instruments. In these circumstances, the correct selection of methods and equipment plays an important role, if the whole work shall be kept within economic limits.

Measuring and control in an individual case.

Measurement can take place :—

1. On the work-piece at the machine tool and on the tool itself during or immediately after production.
2. On the finished wheel, either in the works of the producer or in the receiving department of the user.

In the first case the individual dimensions of the wheel are tested in an endeavour to discover deviations from the correct figures separately, without their influence on each other, and to ascertain the reasons and possible remedies for defects. These measurements can be made from time to time, e.g., after the change-over of the machine tool when starting a new batch of wheels, or on each wheel, for thickness or width of the teeth. If it is necessary to obtain complete information on the state of the wheel, the following must be tested :—

1. Form of the tooth. (*a*) Exact involute gearing, including surface of the flank. (*b*) Bottom diameter (angle of action).
2. Pitch (individual and summary deviation). (*a*) Chordal pitch, and (*b*) circular pitch.
3. True running within the spaces.
4. Thickness of tooth and width of space.
5. Angle of obliquity or direction of the tooth.

In the second case, measurement of the finished wheel and the determination of any defects with regard to the function of the wheel is necessary, for example, by a rolling-off process.

TESTING FORM OF TEETH.

A defective tooth form causes an incorrect gear ratio, as well as wrong results in the measurement of pitch, thickness of teeth, true running, etc.

A reliable instrument for *testing the form of the flank* is indispensable for producing toothed wheels. It is recommended that it should be kept separately in a room protected against dust, vibrations and fluctuations of temperature, situated as near as possible to the actual production shop, and a test-curve should be taken of the first wheel of each new batch, not only for ground wheels with small tolerances, but also for those produced by milling or slotting.

Fig. 1 shows an instrument with continuous adjustment of the radius of the base circle. It is fairly complicated and comparatively expensive, but it is almost essential to have at least one of these instruments in shops where the dimensions of the wheels are changing continuously. Instruments with discs for the radii of basic circles also give very satisfactory results, even in the case of precision wheels, if subjected to expert treatment and control. Necessary suppositions are : small tolerances, say, fitting of the discs in the instrument without play ; the pressure between rolling-disc and rolling-ruler remaining equal ; correct alignment of the centre lines of the rolling-disc and the toothed-wheel, and sufficient sensitivity of the recording mechanism.

TESTING SURFACE OF FLANKS.

There is still no instrument that can be recommended for immediate use in the workshop, but one will probably be developed either for testing the tooth-flanks by means of a feeler and enlarging the irregularities of the surface with the help of a copying mechanism, or by additional devices to optical instruments for producing suitable silhouettes. An example of the first alternative

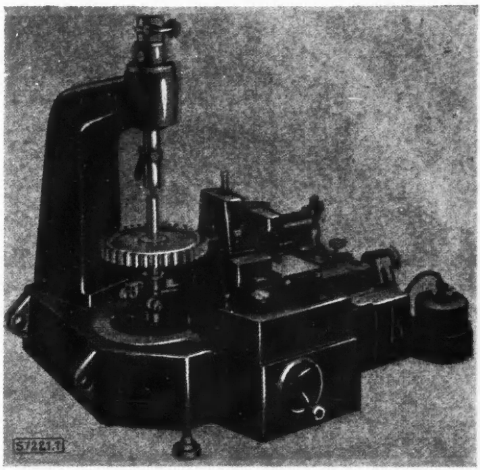


Fig. 1. **Instrument for testing the face line of teeth by continuous adjustment of the base circle radius. Maker ; Carl Zeiss, Jena.**

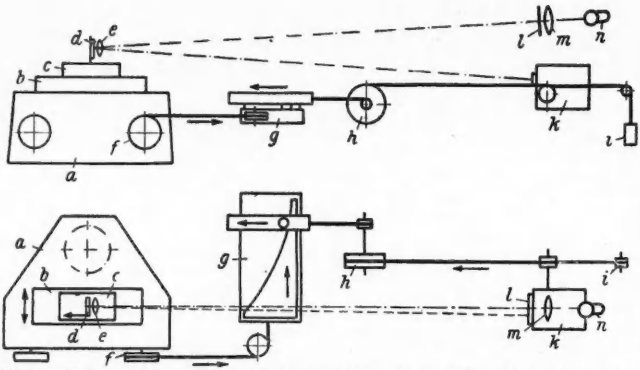


Fig. 2. **Photographic copying of shape and surface of flank faces of teeth according to Dr. K. Bürger.**

(a) Test instrument; (b) Transverse slide; (c) Longitudinal slide; (d) Mirror; (e) Copying lens; (f) Hand wheel of drive; (g) Curves for paper-feed; (h) Intermediate countershaft; (i) Tightening weight; (k) Registering instrument; (l) Lighting slot; (m) Collecting lens; (n) Source of light.

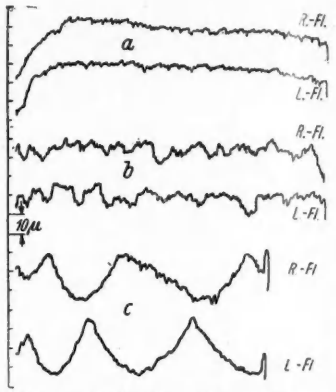


Fig. 3. **Records of surfaces of tooth-flanks produced by the arrangement according to Fig. 2.**

(a) Smooth-ground; (b) Cross-ground; (c) Tooth-flanks milled.
R — Fl. = right flank *L — Fl.* = left flank.

is shown by the experimental arrangement in Fig. 2. The copying on photographic paper shows a thousand-fold enlargement and examples of the curves obtained are given in Fig. 3. There is no doubt that the rounding of the contact-point causes some distortion in the curves, but still the method makes it possible to determine numerically the deviations of the surface of the flank from the correct form, at least approximately.

TESTING CIRCULAR PITCH AND CHORDAL PITCH.

It is not possible to give a concise answer to the often disputed question of whether the chordal pitch *te* or the circular pitch *t* should be tested (Fig. 4).

In the case of *ground* toothed-wheels it is generally sufficient to measure *te* when testing the individual deviations from tooth to tooth. With *milled* wheels, however, the *te* measurement from tooth to tooth is useless, as only the distance of points which are produced by the invariable distance of two edges of a hob cutter is determined by this measurement (see Fig. 4c). In this case the measurement of the circular pitch only is of use.

With *slotted* or *scraped* wheels, the answer is doubtful. Preference should be given to testing the circular pitch as the deviations of the arcs may, in some circumstances be larger than those of the chords. Striking differences between the *te*- and *t*-measurement were obtained on a slotted wheel after lapping (Fig. 5). Whilst the chordal pitch is completely correct, the deviations from the circular pitch remain almost unchanged.

Chordal pitch. Quite a different aspect is obtained, however, if not only the equality of pitch from tooth to tooth is considered, but the absolute magnitude of each individual pitch is taken into account. The difference will be recognized by comparing Fig. 5 with Fig. 6, which shows the deviations of the chordal pitch before and after lapping in absolute figures, i.e., with regard to the measurement of *te* as it should be. The lapping procedure, of course, results in a larger equality of the chordal pitch; the absolute value, however, is changed considerably. From this fact a general conclusion can be drawn as follows: the measurement of the chordal pitch only gives a complete explanation of the real state of a toothed wheel, if the defects are determined in their absolute magnitude. For this test, a hand measuring instrument is, to a large extent, applied in the workshops. The centre of gravity of this instrument, when adjusting it to the gauge, should be at least approximately the same as during the actual measurement of the toothed wheel (Fig. 7).

The usual clock dial micrometer must be replaced by a precision-feeler (Krupp or Mahr, Fig. 8). Fig. 9 shows a construction, developed especially for measuring chordal pitches of model- or master-toothed wheels, where the plane and the bracket support of the instrument are replaced by a roller *a*, which is especially useful for the measurement of *te* of internal teeth.

Circular Pitch. Individual deviations.

Fig. 10 illustrates an instrument for testing individual deviations of circular pitch and should be self-explanatory. It must be taken into account that so long as the wheel to be tested is borne by a spindle through the centre, as is usual for small or medium sized w..eels, the influence of any deviation from true running is included in the measurement, whilst this is generally avoided with the usual methods of measuring the chordal pitch. A measurement taken from the outside circle may include the influence of any eccentricity between this circle and the pitch circle.

Deviation Summary. A correct determination of the summary of deviation from the circular pitch is only possible by means of the measurement of angles, preferably with theodolite and collimator. This method takes a comparatively long time and should be used solely for testing the pitches of gear-cutting machines.

In the case of hobbing and slotting machines, the theodolite is put directly on the machine table or on a

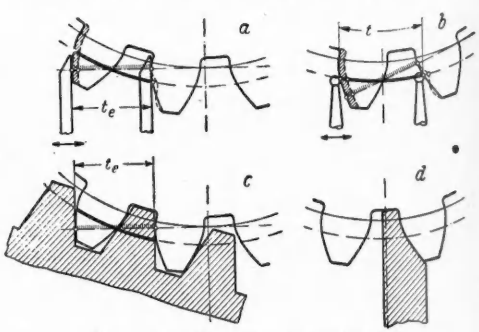


Fig. 4. **Chordal pitch and circular pitch.**

(a) Measurement of chordal pitch *te*; (b) measurement of circular pitch *t*; (c) dependence of the chordal pitch *te* when cutting teeth by means of tools with multiple cutting edges (hob cutters); (d) When grinding the flanks the pitch is independent of the tool.

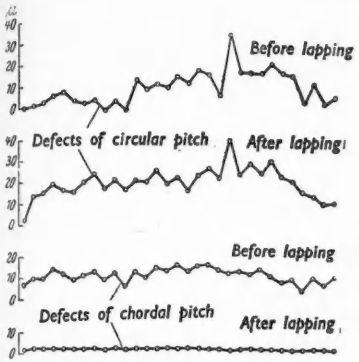


Fig. 5. **Inequalities of circular pitch and chordal pitch before and after lapping.**
Deviations of the circular pitch before and after lapping.

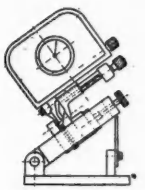


Fig. 7. **Adjusting the hand measuring instrument for testing the chordal pitch on a gage in a sloping position.**
Type : National Institute of Technical Physics, Berlin.

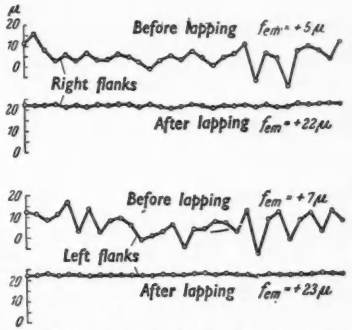


Fig. 6. **Deviations of the chordal pitch before and after lapping in absolute figures.**
fem = mean deviation of the chordal pitch

model-wheel cut by the machine; in the case of grinding machines it is recommended that the index dials should first be tested in a suitable device and then to provide for true running of the dials within the smallest possible limits when inserted into the machine.

True running of the teeth.

The true running of the teeth is largely influenced by the bore of the wheel, the spindle through the bore and the spindle bearings. As each hole has a certain allowance in the diameter, there is theoretically only one spindle running within one bore without any play, and each measuring instrument can be used only for one wheel, the bore of which corresponds to the spindle of the instrument with the smallest possible allowances. This consideration suggests that the use of a cylindrical spindle as part of the support of the wheel being tested should be avoided.

Support. The simplest and, up to now, most reliable part of the support for a wheel in a testing instrument is a mandrel of conical form, tapering less than 1 : 12000, or, in the case of model-or master-wheels, less than 1 : 25000. As hole allowances cannot be bridged by mandrels of so small a taper it is not possible to avoid, for a certain nominal dimension, the application of two, and sometimes three, mandrels with correspondingly fine graduation in the diameters. If expanding mandrels could be developed to a higher stage of perfection than at present, it is possible that they could be used for measuring instruments.

It is important that the centre sockets of the measuring mandrels are carefully finished. Their internal cones should be smoothly ground and if possible lapped afterwards. The centre marks should be spherically rounded off, generally speaking, so that injuries to the centre socket can more easily be avoided.

As the mandrels in spite of artificial ageing of the material, are sometimes distorted after long usage, a test for true running is necessary from time to time. This should be done at various points, at least at either end and in the middle. If a mandrel is required with allowances of 2 to 3 μ for true running, a feeler has to be used for the test without any indication of a tolerance when used in the reversed position. If this is not possible, the feeler must be lifted each time the mandrel has been rotated a bit further on for testing, in order to work always with a feeler coming out and avoiding eventual play.

As the truing allowances of gear wheels are generally very small—usually only several hundredths of a millimeter, and even down to 10μ—the usual clock dial micrometers should not be used, but more sensitive precision feelers applied instead. It is more economical to use a more accurate testing instrument, as the allowances for the workshop can be increased if the uncertainty of measuring is decreased.

With regard to testing for truing, it is recommended that the test piece be placed vertically between the centres in order to guarantee symmetrical loading at the centre hole.

Cylindrical bolts have stood the test as size blocks,

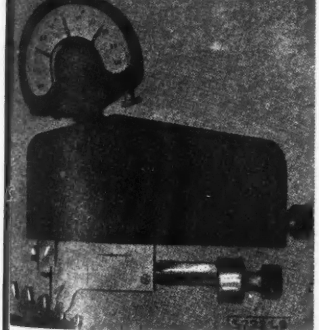


Fig. 8. **Modern testing instrument for measuring the chordal pitch. Construction: Carl Mahr, Esslingen a.N., fitted with precision indicator "Klein-Millimess." The instrument is also provided with a supporting roller.**

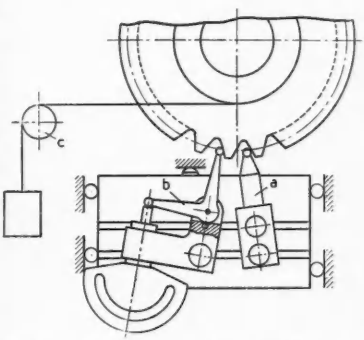


Fig. 10. **Testing of circular pitch with fixed stop and mobile precision feeler.**
(a) Fixed stop; (b) Mobile precision feeler; (c) Weight producing a continuous and equal pressure of the wheel against the stop (a).

Fig. 9. **Hand instrument for measuring the chordal pitch and fitted with a supporting roller. Construction: National Institute of Technical Physics, Berlin.**
(a) Supporting roller; (b) preci ion indicator "Mikrokator."

e.g., in the form of standard plug gauges, which are suspended by a thread and inserted into the spaces one after another. The reversing point of the hand of the measuring instrument is then read off, while the toothed wheel is rotated slowly.

What may best be considered as a truing test can be summarized as follows: In all cases where specially small allowances are unnecessary (perhaps below 20μ), such a test is required only for spaces at more or less large distances. This lengthy measuring method can usually be replaced by rolling the test piece on a master wheel in a two-flank instrument, using perhaps a run of ball- or roller-shaped teeth or a worm. It is, however, possible to keep the truing defects of the finished wheel within the permissible limits and to save a large part of the testing work, by accurate preparatory work on the wheel, especially so far as the holes are concerned, and by continuous supervision of the clamping and centring parts of the machines.

THICKNESS AND WIDTH OF TEETH, AND WIDTH OF SPACE.

Of great importance for the fitting of wheels in a gearing is the backlash, which again is largely influenced by any truing defects, by the allowance in the centre distance, and particularly by the variation of thickness in the teeth. It is, therefore, necessary to fix clear allowances in the thickness of the teeth for the production of the individual wheel. The dimensions depend on the feed of the tool and the thickness of each individual tooth must be measured before the wheel is unclamped and taken off the machine. For this purpose the measurement of the width of the teeth is generally carried out in the workshops, and is usually done by means of special calipers which are set according to a parallel precision gauge block, or, better still, to an end gauge with cylindrical measuring surface (Fig. 11).

The accuracy of measuring the width of the teeth may also be increased by the use of special instruments equipped with sensitive precision feelers, as the feeler pressure always remains the same, and is independent of manipulation (see Fig. 12). The measurement of the thickness and of the width of the teeth can be determined by measuring the space in the pitch circle by means of inserted rollers or balls. This method is applied especially to internal teeth.

PARALLELISM OF THE TEETH.

In spur gears the testing for parallelism of the flanks to the centre line is still rare in practice. The marking of high spots on the flanks by rolling the wheel against the master-wheel or against its counterpart in the gear, is generally considered sufficient for drawing conclusions from the appearance of the bearing surface. Sensitive as this method may be, it gives no numerical results and is very uneconomical.

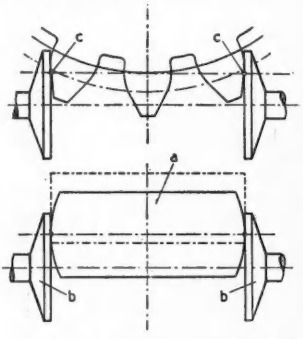


Fig. 11. **Measuring width of the teeth. The setting gauge *a* should contact the measuring surfaces *b* as far as possible at the same points on the flanks of the teeth.**

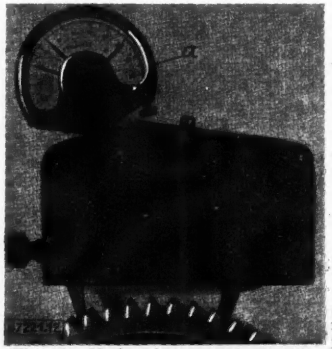


Fig. 12. **Modern instrument measuring the width of the teeth. Maker: Carl Mahr, Esslingen a,N,**

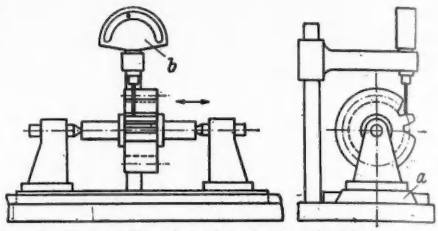


Fig. 13. **Testing direction of teeth of a wheel by means of a feeler.**
The wheel is taken between centres on a cross slide *a*, moving longitudinally while the dial gauge and feeler *b*, copying the surface of the flank, is fixed.

Reliable and numerical results can be obtained only by contacting each tooth-flank separately by a feeler, which is best fixed, whilst the toothed wheel, placed on centres, is moved longitudinally (see Fig. 13).

Testing the direction of teeth in an angular position is of special importance as the advantages of this more difficult and more expensive design can be obtained only if the angles of the two wheels working together are the same within very small limits, i.e., if the two flanks lie really close to each other across the whole width of the wheel. Suitable measuring instruments do not exist in most cases and it is highly desirable, in the future, to remedy these defects as far as possible. As they depend almost exclusively on the setting of the machine tool producing the wheels, it will be sufficient to repeat this test after a batch of wheels has been produced, to make sure that the setting is still correct. The actual measuring work is, therefore reduced to a minimum.

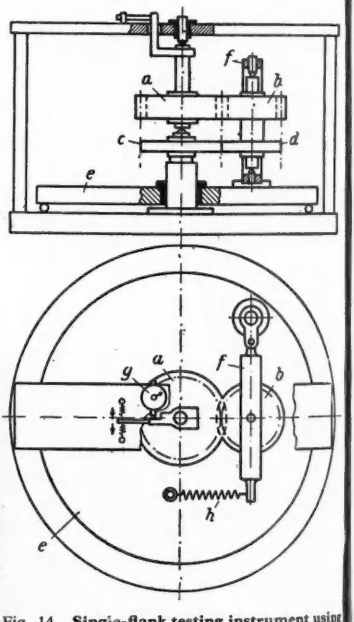


Fig. 14. **Single-flank testing instrument using the rolling-off method.**
Type: Dr. K. Bürger, N.I.P.T., Berlin.
(a) Master Wheel; (b) Wheel to be tested; (c) Fixed friction-disc; (d) Friction-disc connected with (b); (e) Swivelling table; (f) Swinging link, bearing wheel (b) and disc (d); (g) Precision feeler and writing implement showing rolling-off defects; (h) Spring pressing the friction-disc against each other

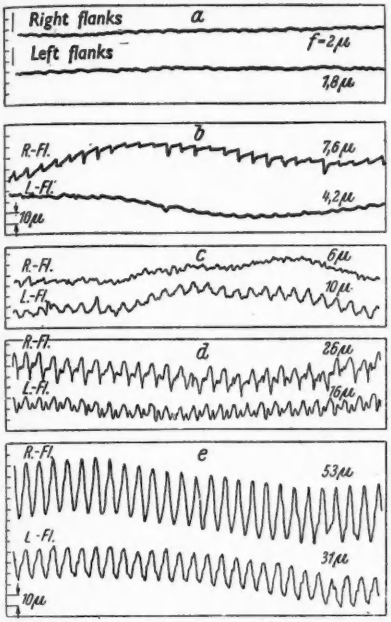


Fig. 15. **Diagrams taken by means of the instrument (Fig. 14), showing the testing of single-flanks and their rolling-off defects originating from various methods of production.** (a) Co-operation of two master-wheels ; (b) Ground wheel and master wheel working together ; (c) Slotted wheel and master wheel working together ; (d) Milled wheel and master wheel working together ; (e) Milled wheel showing large defects, working together with a master wheel.

TESTING WITH REGARD TO COLLECTIVE DEFECTS.

In contrast to testing with regard to individual defects in the workshops as explained above and which requires a considerable amount of time the test by rolling the workpiece against a master will be used more and more in future as an acceptance test.

The generally approved double-flank testing method shows so many fundamental defects that it should be supplemented by the single-flank method. As, during the former test, the two wheels run without any play, i.e., right and left flanks mesh at the same time, but it does not reflect the state of affairs during the actual working of the gears, and must be considered only as an approximate method of judging the quality of the gearing. It is, of course, sufficient if only the true running and variations in thickness of teeth are controlled. The testing previously explained with insertion of balls or rollers, deserves preference, because of the considerably shorter time required.

The single-flank method combines the wheel to be tested with a mate on the same centers on which it has to work afterwards, i.e., with the normal flank-clearance. Thus, it compares the turning motion to be tested, with a faultless friction disc gear. The result of this test is the difference in the angular velocities originating from defective gearing. In spite of the fact that this method alone, as developed by Saurer, can be considered as a really correct test of the total defect, it is rarely used in practice as suitable instruments have, up to now, been lacking. Investigations have been made in the National Institute of Technical Physics by means of a new single-flank testing instrument (Fig. 14). This shows that in the majority of cases the defects in the two flanks are very different with regard to magnitude, as well as to course (Fig. 15). It is, therefore,

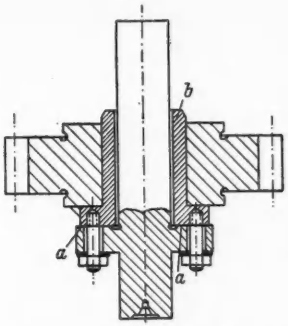


Fig. 16. **Special mandrel with adjustable sleeve.**

absolutely necessary to pay more attention in future to the single-flank test method and to develop instruments for its use in the workshops.

Rolling-off gages.

It is only possible to consider testing the total defect, for the exclusive approval of a toothed wheel, if a sufficient number of rolling-off gauges are at hand. In this connection master wheels are of decisive influence. A few years ago it was the general opinion that these master wheels could not be produced with the necessary high degree of accuracy ; but modern developments in production have changed this point of view completely.

Master gear wheels of medium module can be produced almost without defects with regard to the tooth outline. As far as the chordal pitch is concerned variations of $\pm 2\mu$ and pitch steps of 1 to 1.5 μ from tooth to tooth can be obtained ; but truing defects still give considerable difficulty when producing master wheels. The special mandrel (Fig. 16), however, provides a means (shown in Fig. 17) of decreasing the truing defects of master wheels to below 5μ. The problem of producing master wheels with small modules below 1 mm. has still to be solved ; but it is expected that in this case too the difficulties will not be insurmountable. The development of standards for these master toothed wheels has, in fact, already started.

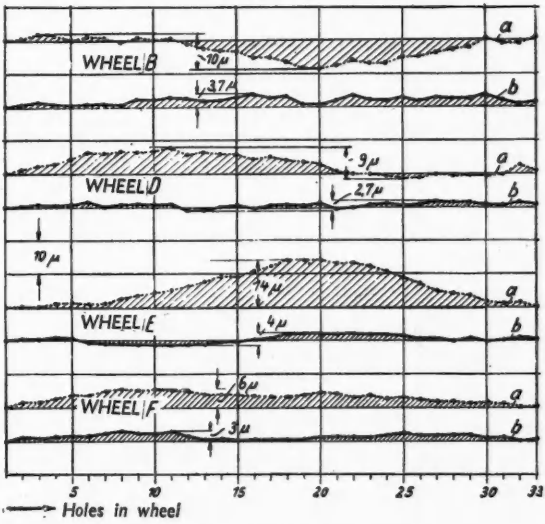


Fig. 17. **Truing defects of master wheels.** (a) Represents measurement taken directly from the hole in the wheel ; (b) Shows the measurement when using the special mandrel according to Fig. 16.

THE PRODUCTION OF GRAY IRON CASTINGS IN METAL MOULDS.

By C. ENGLISCH. (From *Die Giesserei*, Vol. 30, No. 16/17, August, 1943 ; pp. 181–189.)

THE employment of iron moulds for the production of gray iron castings is not yet very frequent, and is therefore little known. Very often this process is apt to be looked upon with a certain distrust, but this may be due to the fact that successful application of this process requires a considerable amount of experience in order to be able to overcome the many inherent difficulties ; while those in possession of such experience have often been unwilling to communicate it to others.

From the economic point of view, the use of metal moulds can be justified only where large numbers of identical castings are to be produced ; while from the technological point of view its application can be justified only where the high quality of these castings, that is density and accuracy of dimensions, is really called for. In the manufacture of engines, automotive vehicles, and of electric motors, there exist many cases where the latter requirements actually prevail, particularly in connection with the manufacture of large numbers of standardised components.

The experiences related below, which refer solely to the use of stationary moulds, were made in the foundry of a maker of hydraulic and pneumatic brakes and of refrigerating machinery, whose annual production amounted to some two million gray iron castings made in metal moulds.

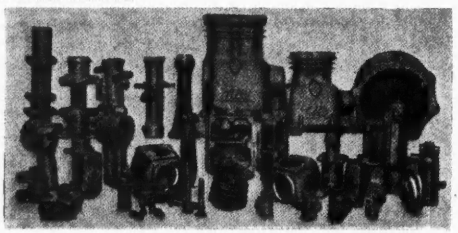


Fig. 1. **Typical moulds.**

In the design of pieces to be cast in metal moulds, attention must be paid to the special characteristics of the material to be used in this process and its solidifying characteristics. Castings of this type must also be so designed as to ensure ease of machining. As illustrated in Fig. 1, the use of iron moulds extends from very small pieces weighing some 70 grams to pieces weighing as much as 50 kg. Among these castings are component parts of hydraulic plants which have to be tested at pressures as high as 300 atm. g. Other parts produced in this way are cylinders and cylinder covers of refrigerator compressors and air compressors, cylinders with cast-iron cooling fins and other fairly complicated pieces. Valve-seats and valve tappets for internal combustion engines are also produced by this method. Wall thicknesses of the castings produced range from 3 to 20 mm.

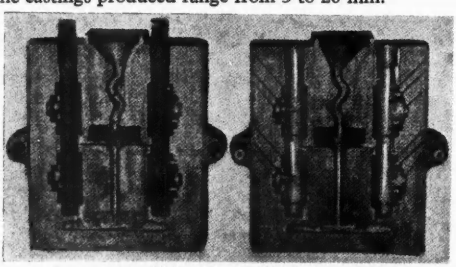


Fig. 2a. **Mould for 4 hydraulic brake cylinders.**

As will be seen from Figs. 2a, b, and 3a, b, bottom pouring is usually employed. Care must be taken that when pouring, the flow of the liquid metal in the mould is uniform and smooth. Turbulence must be avoided, and any direct impingement of the liquid metal upon mould surfaces must be prevented by appropriate guidance of the flow. Inclusions and dirt must be excluded by the provision of adequately arranged gates and risers. Owing to the non-porosity of the mould

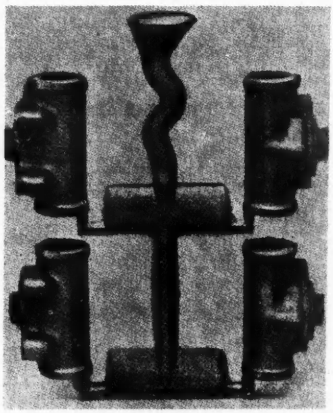


Fig. 2b. **Casting produced from mould Fig. 2a.**

the provision of a sufficient number of vents is particularly important, but these must not be unduly large, as this would lead to the formation of fins and other projections impeding the removal of the casting from the mould.

Care must also be taken to ensure sufficient cooling of all parts of the mould in order to avoid hot spots. The local wall thickness of the mould must therefore be dimensioned from the aspect of uniform cooling. Where necessary, the external surfaces should be provided with special cooling fins. At the same time the mould must be made sufficiently rigid by the provision of external ribs, as shown in Fig. 4, in order to prevent deformation by thermal expansion stress.

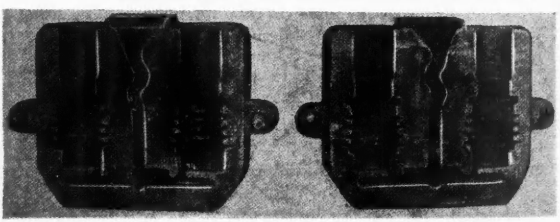


Fig. 3a. **Mould for 2 air compressor cylinders.**

The cores—as far as they are required—are preferably made of sand ; and they must be properly blackened. Where necessary the cores are held in place by chaplets which will be melted and absorbed by the liquid metal. Small cast iron cores find occasional application where they can be attached to the mould and where their efficient cooling can be ensured. Cores of this type can be seen in Fig. 5, which shows the mould for the cylinder head of a small compressor. In this case the core which represents the compression space, is affixed to one-half of the mould. Because of the high

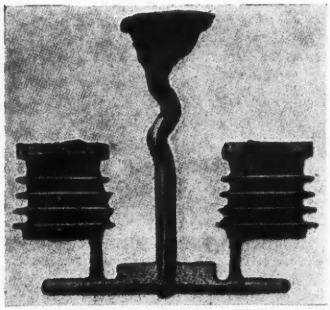


Fig. 3b. **Casting from mould Fig. 3a.**

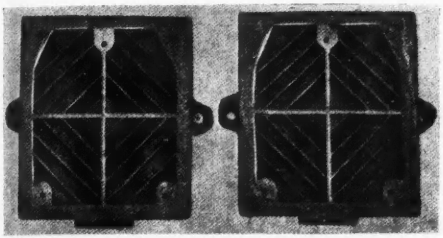


Fig. 4. **Strengthening ribs on back of mould.**

temperature to which cast iron cores are exposed, they are subject to considerable wear and should therefore be replaceable.

For pouring, the moulds are bolted to pouring stands with a self-arresting lever-operated sliding device for opening and closing the mould, as shown in Fig. 6. Preparation of the moulds for pouring includes their swabbing with a watery solution of waterglass and clay or graphite. This mould-wash is applied after heating and special care must be taken to thoroughly swab all parts where temperatures will be highest. In the case of small and simple pieces cast solid, the mould-wash is sometimes omitted. On the average, one mould-wash will last through 50 to 60 or even 100 casting cycles. Its main function is to prevent unduly rapid heating of the mould which might cause distortion. If it is not properly prepared and applied, the mould-wash will flake off and cause faulty castings.

Prior to making a casting, the mould is cleaned by air-blast, and followed by blackening by means of an acetylene flame. The first few castings made while the mould is still cold are discarded. Production proper is commenced when the mould has reached a temperature of 220 deg. C. It is important that this temperature is constantly maintained on the mould face. Cooling is carried out by means of a stream of air discharged from a nozzle against the external front and rear surfaces.

Pouring is carried out at the lowest possible temperature, taking into account size and wall thickness of the casting to be produced. Small thin-walled pieces are cast at approx. 1400 deg. C. and large pieces with thick walls at about 1320–1300 deg. C. Even so, considerable quantities of heat must be removed from the mould by cooling. Where small pieces of no more than 4 kg. in weight are involved, a casting is produced approximately every six minutes. In the case of larger pieces, a casting is produced every 12, 18 or 24 minutes. According to the size of the piece, the casting is removed some 15–30 seconds after pouring. The open mould is cleaned with compressed air and blackened again. After putting the cores in place, the mould is closed and is ready for pouring. In the foundry (Fig. 7) 50 moulds, occupying a floor space of 300 sq. m., are in operation, the daily production amounting to 5.5 to 6.0 tons of castings. The standard composition of the castings is 3.3 per cent total C, 2.5 per cent Si, 0.5 per cent Mn, 0.5 per cent P, and about 0.1 per cent S.

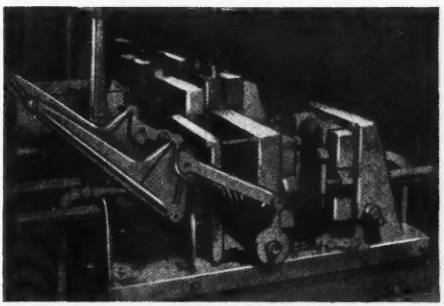


Fig. 6. **Casting device.**

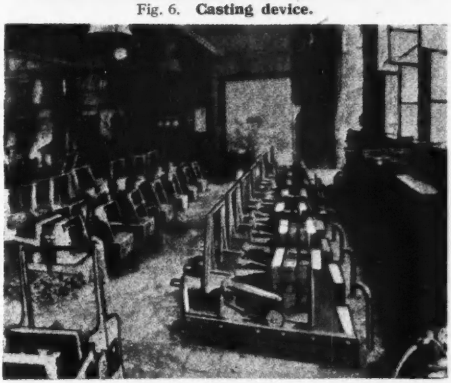


Fig. 7. **Foundry shop using iron moulds.**

Thin-walled castings show a mottled structure, while in thick-walled pieces the interior is pearlitic, with surfaces and edges showing cementite. Annealing is therefore required. This is carried out in muffles held at 900 deg. C. for two hours in a gas fired furnace. Upon cooling to 300 deg. C., the muffles are removed from the furnace and the castings taken out. Their

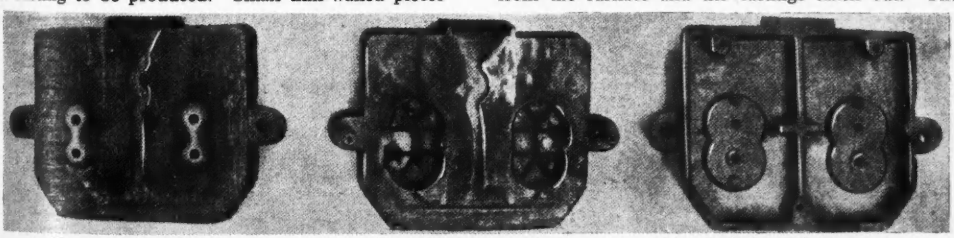


Fig. 5. **(a) Mould for 2 cylinder covers of refrigerator. (b) Compressor with attached cast iron cover.**

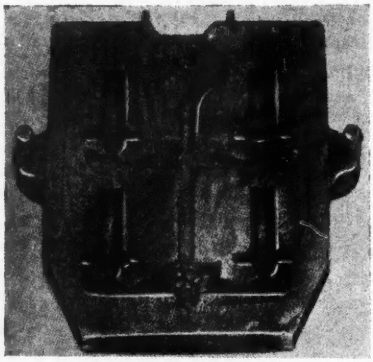


Fig. 8. **Worn-out mould.**

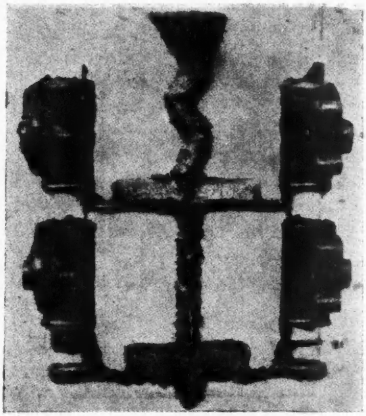


Fig. 9. **Casting produced from a worn-out mould.**

final cooling takes place in air. It is interesting to note that the structure of the annealed castings is satisfactory for uses where the piece as e.g., a cylinder for a small compressor is subject to moderate wear. As a matter of fact, such cylinders have given up to ten years satisfactory service without showing much wear.

The hardness of the annealed pieces approximates to 150–180 Brinell, and their tensile strength is of the order of 22–26 kg. per sq. mm. according to wall thickness. For special purposes alloyed castings are produced, as for instance, valve seats of Cr-Mo or Cr-V material. Their heat treatment is identical with that of the non-alloyed material. Where especially high wear resistance is required, non-alloyed castings can be hardened to 50–54 Rockwell by quenching in oil. The martensitic structure of hardened pieces can be changed by tempering into a martensitic-sorbitic one. Hardened and tempered castings are used for valve-seats, valve-guides, and valve-tappets, and also for bearing bushes of minor importance.

Faulty castings may be caused by a number of mistakes made by the operating personnel. Thus, for instance excessive blackening will produce depressions in the surface of the casting. Improperly dried or insufficiently ventilated cores will result in castings with porous surfaces. If the moulds are not tightly closed, runouts will occur at the joints. Delayed or careless ejection of the casting from the mould may lead to fissures or fracture of the latter. Faulty packing of the castings in the muffles or a wrongly adjusted furnace atmosphere in the annealing treatment will cause surface decarburisation. Decarburised, ferritic surface layers are of great toughness and cause difficulties in machining the piece. An insufficient anneal caused by unduly low furnace temperature or premature termination of the annealing treatment will result in hard surfaces and exposed parts, making machining impossible.

The service life of a mould made of cast-iron with special alloying additions will depend upon size and wall thickness of the casting produced. In the founding of simple pieces, considerably more than 10,000 castings can be produced with a single mould, while in the case of larger and complicated pieces, 3,000 castings will be an average figure. The service life of a mould is terminated by burning out at the spots exposed to maximum metal temperature. A mould in such a condition is illustrated in Fig. 8, while a casting produced from a mould deteriorated by burning-out is shown in Fig. 9.

TECHNICAL INNOVATIONS IN THE SOVIET TANK INDUSTRY.

By Jacob Karpov.

We have received the following letter from the Press Department of the Soviet Embassy :

Dear Sir,

The attached article has just been cabled from Moscow and we have pleasure in sending it to you for exclusive publication in your journal.

Information on Russian engineering developments has been scarce since Germany declared war on the Soviet Union. Russian technical books and periodicals have been almost unobtainable during the last four years. In the years 1940 and 1941 a number of articles appeared in The Engineers' Digest which, abstracted from current Russian technical journals, have recorded some of the important achievements of Russian science and industry. Since June, 1941, however, all our efforts to obtain Russian engineering literature have been in vain.

We are glad to publish the article on technical innovations in the Soviet tank industry, despite the fact that the article does not contain any descriptions and technical details in which our Readers will no doubt be most interested. On receiving the manuscript we approached the Soviet Embassy for detailed articles on the process of manufacturing almost finished cast tools, and on the chemico-mechanical method of grinding mentioned in the article. The Press Department of the Soviet Embassy has promised us most obligingly to cable for the articles requested and we are pleased to announce that we shall be able to publish other articles also on recent Russian engineering research and developments in early issues of The Engineers' Digest.

The Soviet T. 34 and K.V. tanks, which were a terror to the enemy in the early period of the war, have been constantly improved upon. Soviet engineers, technicians, foremen and Stakanovite workers are also constantly improving methods of work, shortening the production cycle and speeding up the assembly of tanks. For instance, at one big tank works the average time required for the production of a tank was reduced by 60 per cent and at two other works by 66 per cent.

Another innovation is the automatic welding conveyor. Academician Paton's method of automatic welding was introduced only 2 years ago. To-day more than 30 per cent of all tank welding is done by automatic machinery, and is now not only employed for straight seams but also for round and other shapes. While effecting a four to five-fold increase in labour productivity, automatic welding reduces the expenditure of electric power by more than 30 per cent, and that of electrodes by nearly one-third.

An interesting and promising technical innovation which has already become widely popular in Soviet Industry is the production of cast tools. Initiated at a Urals machine building plant, this method makes it possible to cast almost finished cutters, counter-bores and drills. The production of cast tools has now been introduced in many other plants. Cutting blades are also cast and first results show that they are even superior in quality to machined blades.

Improvements have also been made in the grinding of tools. Special grinding wheels are needed for the grinding of tools made of hard surfacing alloys. One of the scientific laboratories in the tank industry developed a chemico-mechanical method of grinding.

Successful results have also been obtained in tank works in prolonging the life of cutting tools. A Urals' machinery plant, introduced nitro-cementation of high-speed steel-cutting tools, which increased their durability by 1.5 and double. Improvements in technological processes and methods of work have been effected in every field of tank production. Thousands of workers are engaged as inspectors of technical control departments. With the aid of gauges and testing instruments they inspect the products, sort them, and test their quality. The question of improved technical control with less labour power employed has long engaged the minds of Soviet engineers. Their main attention was concentrated on getting automatic control. The first steps in this direction have been made in one tank factory which built an automat for testing the quality of bearing pins, and sorts them according to size. This machine replaced the labour of ten inspectors. The designers are now working on automatic machinery for testing other equipment.

TECHNICAL NEWS

Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.

Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

- ✦ NEW EQUIPMENT
- ✦ BOOKS AND CATALOGUES
- ✦ BUSINESS CHANGES
- ✦ PERSONAL
- ✦ NOTES

PERSONAL

Mr. Wilfrid Ayre, chairman of the Burntisland Shipbuilding Co. Ltd., and managing director of Messrs. Hall, Russell & Co. Ltd., shipbuilders and marine engineers, Aberdeen, has been elected a director of Messrs. R. and W. Hawthorn, Leslie & Co. Ltd.

Dr. W. A. J. Chapman, M.Sc. (Eng.), M.I.Mech.E., has been appointed Principal of the County Technical College, Stafford.

Commander Sir Charles Craven, Bt., R.N., chairman and managing director of the English Steel Corporation Ltd., Sheffield, is relinquishing the office of managing director, and **Sir Alexander Dunbar** is succeeding him in this capacity.

Dr. Edwin Gregory, M.Sc., M.I.E.I., F.R.I.C., has been appointed chief metallurgist of Messrs. Edgar Allen & Co. Ltd., Sheffield, in succession of the late Mr. S. J. Hewitt.

Mr. E. Ransom Harrison has been elected a director of Walter Spencer & Co. Ltd., Crescent Steelworks, Sheffield.

Mr. J. H. Hart of the Electrical Measuring Instrument Department of Elliott Brothers (London) Ltd., died suddenly on September 7th, after just completing 60 years continuous service with the firm.

Sir George Jessel, Bt., M.C., has been elected director of Edmundson's Electricity Corporation Ltd.

Mr. John Cyril Jones, B.Sc. (Eng.) (Lond.), A.M.Inst.C.E., A.M.I.Mech.E., Principal, Royal Technical College, Salford, has been appointed Director of Education at the Polytechnic, Regent Street, London, W.1, in succession to Mr. D. Humphrey, B.A., B.Sc., who retired on September 30th.

Mr. C. D. H. Macartney-Filgate has been elected a director of Tube Investments Ltd.

Mr. P. W. McGuire, M.I.Mech.E., has been appointed managing director of the Chesterfield Tube Co. Ltd.

Sir Archibald McKinstry, B.A., M.Sc., M.I.Mech.E., deputy chairman and managing director of Messrs. Babcock and Wilcox Ltd., is retiring from the managing directorship, at his own request, on December 31st. He will, however, remain a member of the board and deputy chairman. **Mr. C. K. F. Hague, M.I.Mech.E.,** the deputy managing director, has been appointed to succeed Sir Archibald as managing director.

Major H. J. W. Marsh, a director of Marsh Pros. & Co. Ltd., Steel Manufacturers, Sheffield, has been killed in action in Italy. He was also director of the Effingham Steel Works Ltd., and Spear and Jackson Ltd.

Mr. Henry Martin, superintendent of Dorman, Long & Co. Ltd., has died at the age of 53.

Mr. J. F. C. Morden, B.Sc., has been appointed chief metallurgist to Bound Brook Bearings (G.B.), Ltd., Birmingham.

Mr. John Neill, director and general manager of The North Eastern Marine Engineering Co. (1938) Ltd., Wallsend-on-Tyne, was re-elected chairman of the National Association of Marine Enginebuilders for 1944-45. **Mr. Allan Stevenson, C.B.E.,** director of Messrs. David Rowan & Co. Ltd., Glasgow, was re-elected vice-chairman.

Mr. A. E. Osborn has been appointed general sales manager of a new division for industrial products, for the Dunlop Rubber Co. Ltd., with headquarters in London.

Sir Felix J. C. Pole has been elected chairman of Ferguson, Pailin Ltd., and **Mr. H. C. Pierson** and **J. S. Ramsden** have been elected directors.

Mr. Sydeney H. Rawlings, founder and joint managing director of The Automatic Coil Winder and Electrical Equipment Co. Ltd., died on September 17th at the age of 64.

Mr. J. E. Richardson, Principal of Oldham Municipal Technical College since 1942, has been appointed Principal of the Salford Royal Technical College.

Mr. C. E. Sexton, M.I.E.E., has been elected a director of the Kalgorlie Electric Power and Lighting Corporation Ltd.

Mr. Thomas R. Shaw, M.I.Mech.E., chief draughtsman of The Churchill Machine Tool Co. Ltd., Broadheath, near Manchester, has died recently.

Mr. W. W. F. Shepherd has been appointed chairman of Messrs. Turner and Newall Ltd. **Sir Samuel Turner,** who has been chairman of the Company for the past 15 years, has agreed to remain on the board and to serve as deputy chairman.

Mr. F. R. Stagg, M.I.Struct.E., has been appointed joint managing director of Thos. W. Ward Ltd., Sheffield. **Mr. H. W. Secker** has been appointed assistant managing director.

Brigadier R. S. G. Stokes, C.B.E., D.S.O., M.C., has been appointed assistant managing director of the Central Mining and Investment Corporation Ltd.

Mr. W. H. Webster has retired from his post of manager of the spring department of the English Steel Corporation Ltd., Sheffield, but will continue to serve the firm in a consultative capacity.

Sir Arthur Winder is retiring from business and has resigned his position as director and general manager of the English Steel Corporation Ltd., Darlington Forge Ltd., and Industrial Steels Ltd. His successor is to be **Mr. F. Pickworth,** who is a director of these companies.

HEAT TREATMENT SPECIALISTS

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THE INSTITUTION OF MECHANICAL ENGINEERS

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FRIDAY, 20th October, at 5.30 p.m. General Meeting. Address by the President, Harry R. Ricardo, LL.D., B.A., F.R.S. "Applied Research."

FRIDAY, 27th October, at 5.30 p.m. Informal Meeting. "The Influence of Engineering on Social Advancement," to be introduced by Edward Reeve, M.I.Mech.E.

FRIDAY, 3rd November, at 5.30 p.m. "Stresses by Analysis and Experiment," by Professor A. J. Sutton Pippard, M.B.E., D.Sc.

FRIDAY, 17th November, at 5.30 p.m. The Thirty-first Thomas Hawksley Lecture, "Research and Development in Aeronautics," by H. E. Wimperis, C.B., C.B.E., M.A., D.Eng.

FRIDAY, 24th November, at 5.30 p.m. Extra General Meeting. "Drawing Office Practice in Relation to Interchangeable Components," by C. A. Gladman, B.Sc. (Eng.)., A.M.I.Mech.E.

THE IRON AND STEEL INSTITUTE

4, Grosvenor Gardens, London, S.W.1.

THE Scheme for National Certificates in Metallurgy in which The Iron and Steel Institute, the Institution of Mining and Metallurgy, and The Institute of Metals are co-operating with the Board of Education, will come into operation so that courses will begin in September, 1945, and the first examinations will be held in 1946 for the ordinary certificate.

INSTITUTION OF NAVAL ARCHITECTS

10, Upper Belgrave Street, London, S.W.1.

THE Council of the Institution have made the following awards: Institution of Naval Architects Scholarship in Naval Architecture (1944) to Mr. K. V. Taylor, of H.M. Dockyard, Portsmouth; Vickers-Armstrong Scholarship in Naval Architecture (1944), to Mr. Ralph Downham, of Messrs. Cammel Laird & Co., Birkenhead; and Yarrow Scholarship in Marine Engineering (1944), to Mr. P. R. Christopher, of H.M. Dockyard, Chatham.

THE GAUGE AND TOOL MAKERS' ASSOCIATION

Standbrook House, Old Bond Street, London, W.1.

THE Export Committee of The Gauge and Tool Makers' Association has continued the praiseworthy effort to prepare the field for the re-establishment and development of British export trade in tools, gauges, moulds and dies, etc.

On the 21st September, the representatives of the Committee, Mr. H. Madeley and Captain L. H. Barton, visited the Norwegian, Netherlands, Portuguese, and Swiss Embassies. The object of the discussions was to ascertain each country's requirements of British tools, gauges, etc., and to determine the best means of supplying these needs.

CATALOGUES RECEIVED

ILFORD BOOKLETS

X-ray Darkroom Practice. A most useful 32-page booklet published by Messrs. Ilford Limited, Ilford, deals with the responsibilities of the darkroom worker in the routine care and processing of X-ray films. The principles and rules laid down in the booklet should prove of great assistance in avoiding those faults in radiographic practice which so easily and often mar the end-results.

Lead Screens for Industrial Radiography. Another Ilford leaflet deals with Lead Screens, consisting of a carefully balanced combination of front and back thicknesses of foil made of a lead alloy specially selected for its durability and radiographic efficiency. By applying Lead Screens the improvement in clarity and contrast is such that, apart from a reduction of exposure times up to 50 per cent, they could always be used with advantage where hitherto non-screen exposures have been the normal practice.

Both leaflets can be obtained by writing to Messrs. Ilford Limited, Ilford, London.

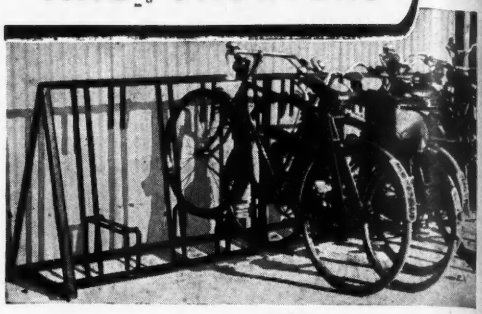
Murex Welding Processes Ltd., Waltham Cross, Herts, describes in leaflet Ref. M. 34 the Murex "Hardex" electrodes designed for making cutting edges equivalent to super-high-speed steels. Leaflet Ref. M. 35 gives details of the Murex "Bronalex" electrodes, suitable for welding complex aluminium bronze alloys.

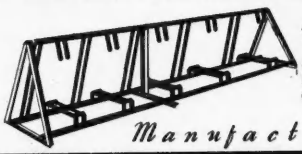
The Midland Iron Co. Ltd., Rotherham, commemorated recently the centenary of the firm. A well illustrated 32-page booklet published for the occasion, gives many interesting details from the history of the Midland Iron Company.

Surfacing and Screw-Cutting Lathes. Henry Broadbent Ltd., Sowerby Bridge, Yorkshire, have issued two 4-page pamphlets describing their 12½ in. and 14 in. centre "M" type high speed, all geared self acting, sliding, surfacing and screw-cutting lathes. The principal dimensions of the machine are as follows:—

Height of centres	14 in.	12½ in.
Admits between centres	7 ft. 0 in.	6 ft. 0 in.
Swing clear of Bed	28 in.	25 in.
Swing clear of Saddle	21½ in.	19 in.
Spindle speeds, 12 changes, r.p.m.	5.8 to 200	6.7 to 230
H.p. required	15	12½
Floor space occupied	16ft. 9in. × 4ft. 11in.	15ft. 0in. × 4ft. 6in.

CYCLE RACKS AND SHELTERS





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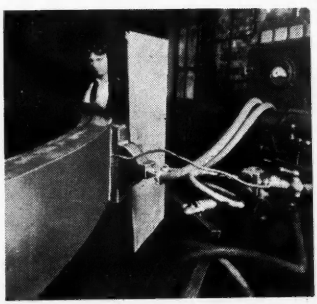

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